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SCREW PROPELLERS.

METHOD OF DESIGNING SCREW PROPELLERS, BASED ON FROUDE'S THEORY OF SCREW PROPELLERS, AS APPLIED BY BARNABY AND ON ACTUAL TRIALS OF VESSELS OVER A MEASURED-MILE COURSE.

BY COMMANDER C. W. DYSON, U. S. N., MEMBER.

In Froude's investigation of the action of screw propellers, and in Barnaby's development of Froude's results, the model propellers used had a constant form of developed area of blade and a constant ratio of greatest width of development to diameter of the propeller.

The form of development used was an ellipse, of which the major axis was equal to the radius of the propeller, and this form of development was used for all values of pitch ratio, $\frac{P}{D}$, that were experimented with.

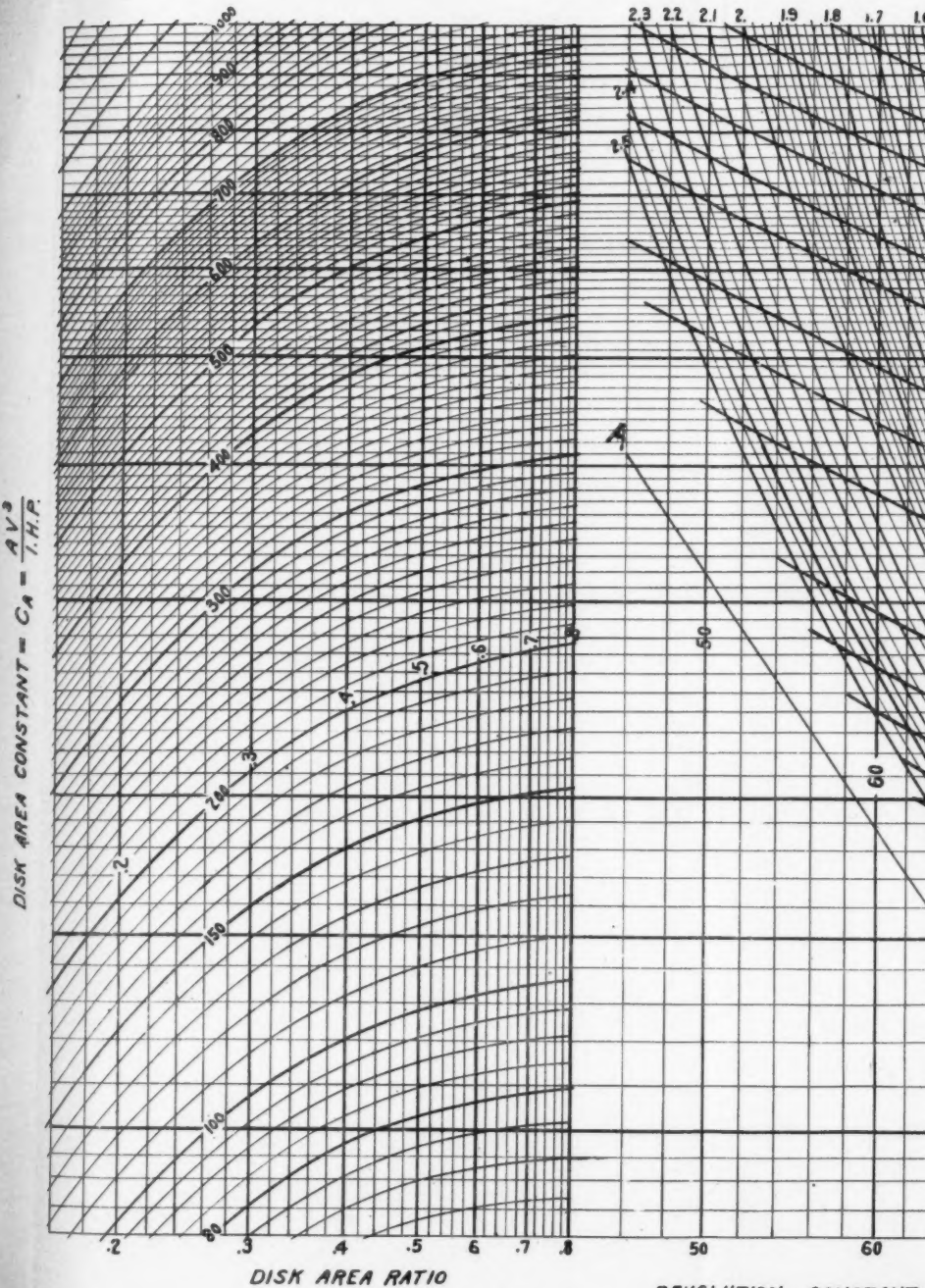
The results obtained from these model tank experiments are correct only so long as we adhere to the ratio of developed area to disc area used by Froude and Barnaby, and to the elliptical form of developed area. We have no way of allowing for increase or decrease in the developed area ratio,

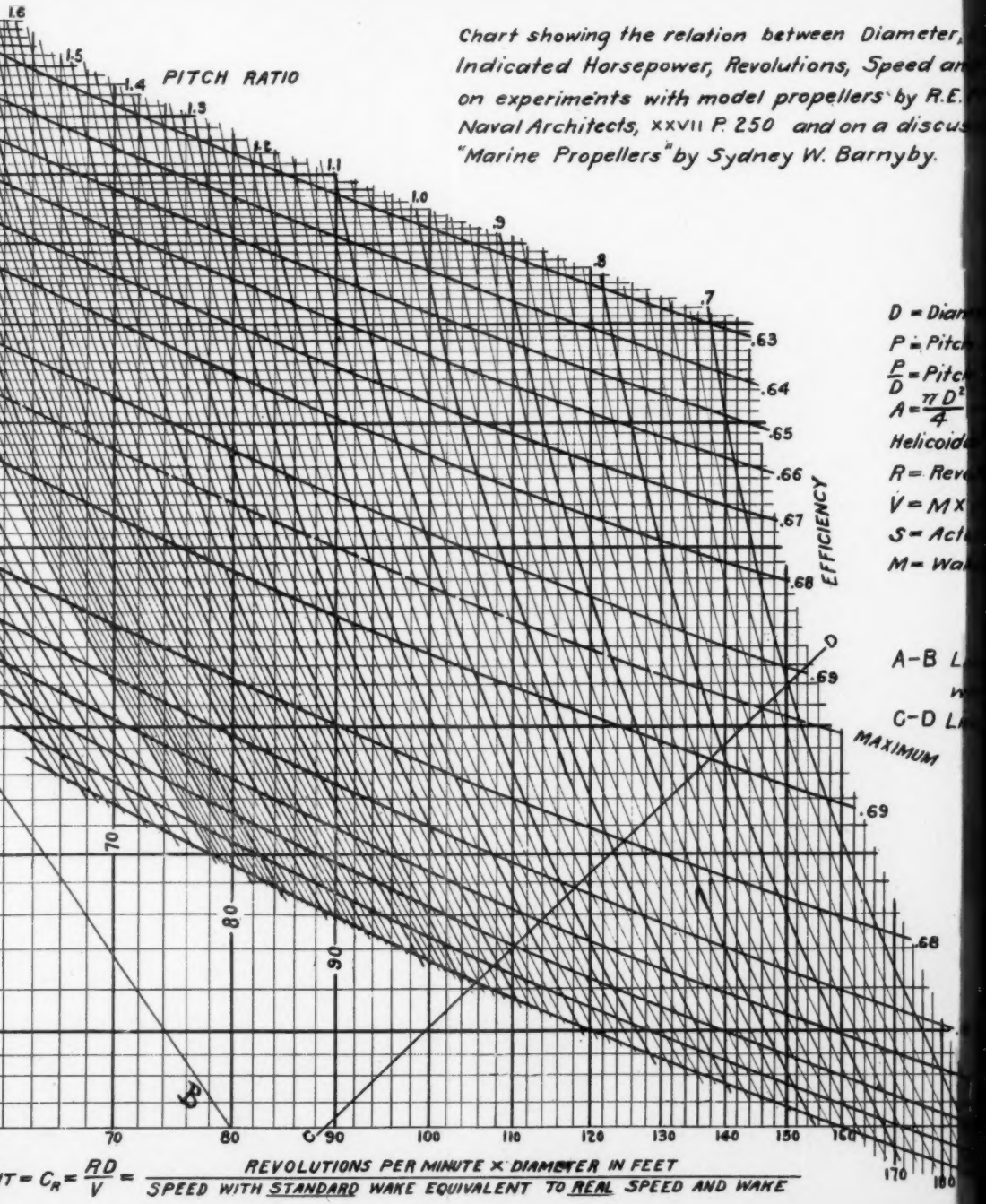
except the rough one of assuming that for the absorption of equal powers the thrust will vary inversely as the developed area.

By investigation of what occurs when we broaden or narrow the elliptical blade from the width used in the experiment, it is readily seen that this method of correcting for variations in developed area ratio is not only crude but is absolutely incorrect, for as the blade widens for any particular pitch ratio the length of its resistance arm increases beyond that of the standard width elliptical blade, and as it narrows, the length of this arm decreases. In the first case, the resistance to turning of the blade is increased, not only by the increase in area of the blade but also by the increase in length of the radius to the center of blade pressure, while in the second case the resistance to turning decreases, not only on account of the decrease in blade area but also on account of the decrease in length of the resistance arm.

If elliptical blades of varying pitch ratios, but having the same area of projection on the disc, be laid down, it will be noted that the entire form of projection, not only on the disc but also in the fore-and-aft plane of the axis of the propeller, changes as we pass from the lower to higher pitch ratios. This same change of form of projected area also occurs if we lay down blades of the same pitch ratio but of different values of developed area ratio, thus in both cases showing that not only the resistance of the blade due to change of surface, and in the first case change of pitch, has been affected, but it has also been further modified by the change in distribution of surface.

In order to guard against these additional effects in the resistance of the blades of propellers caused by changing the developed area ratios, the Bureau of Steam Engineering of the Navy Department practically nullifies some of these effects by using a standard form of projected area. This form for a ratio of projected to disc area of .32, together with the corresponding developments for various ratios of diameter to pitch, is shown on Plate 6. The true form of the development of each blade near the hub is shown in dotted lines, while the


DISK AREA CONSTANT = $C_A = \frac{AV^3}{T.H.P.}$
1000
900
800
700
600
500
400
300
240
200
150
100
80
.2
.3
.4
.5
.6
.7
.8
2.3 2.2 2.1 2. 1.9 1.8 1.7
2.4
2.5
50
60
DISK AREA RATIO
REVOLUTION CONSTANT



$T = C_R = \frac{RD}{V} =$

Plate 1.

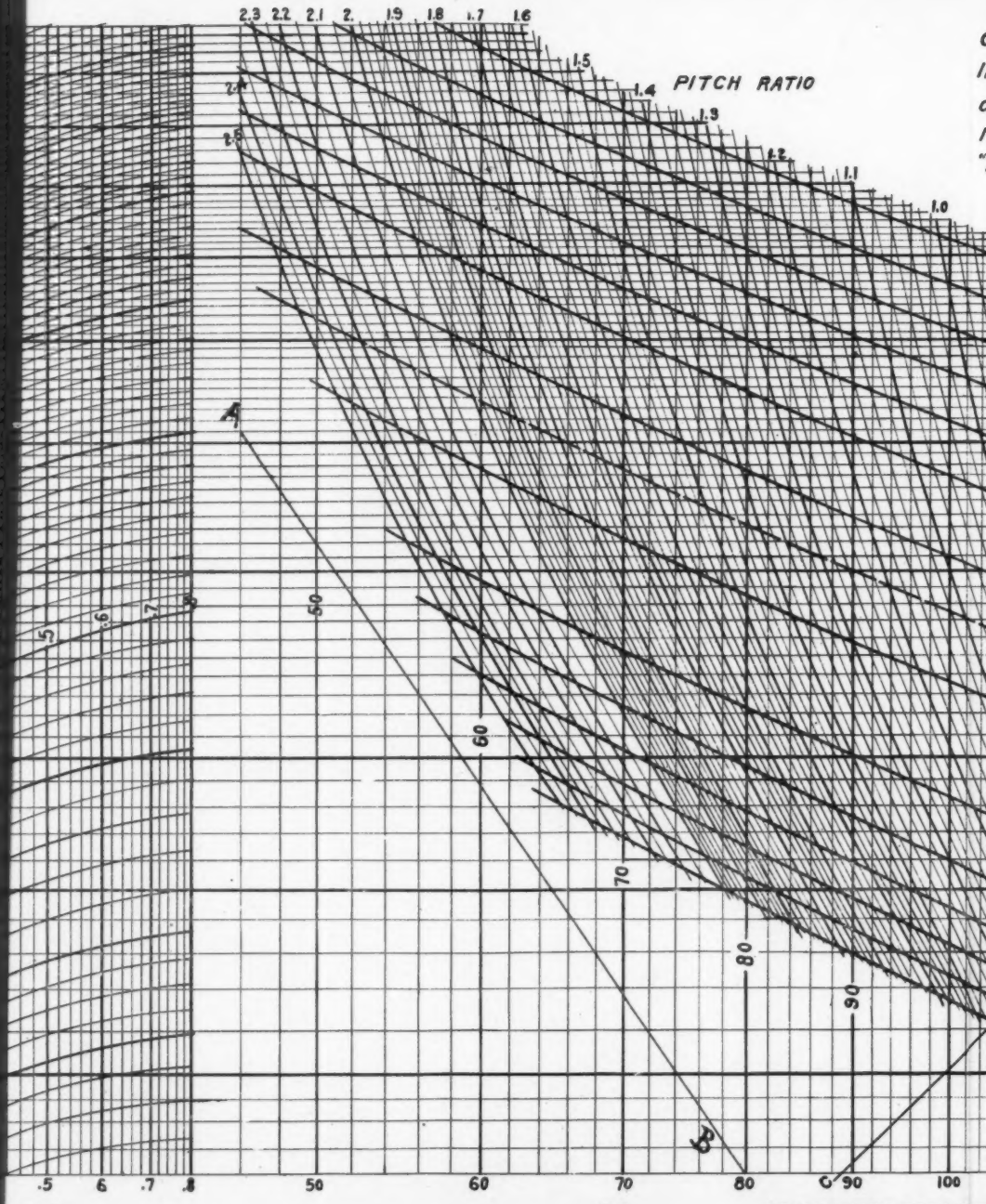
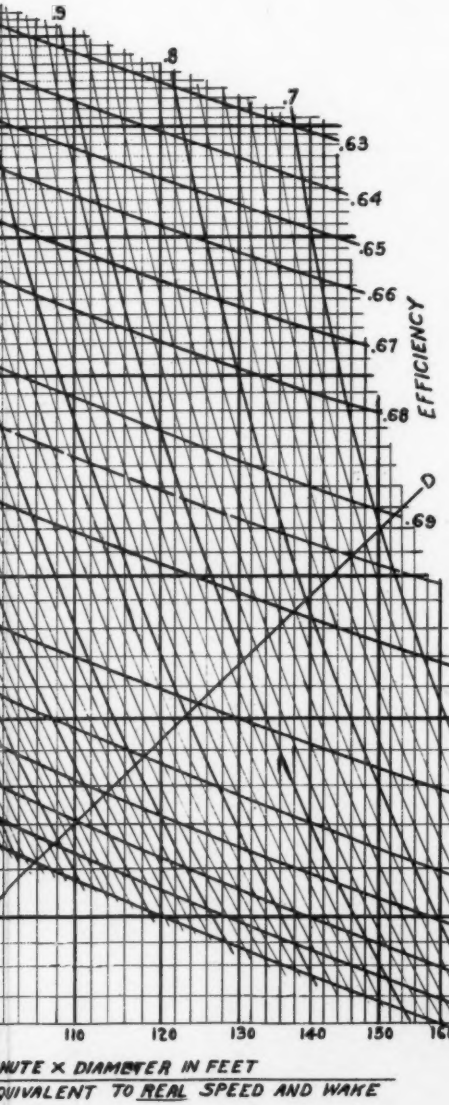

2.3 2.2 2.1 2. 1.9 1.8 1.7 1.6
1.5
1.4
PITCH RATIO
1.3
1.2
1.1
1.0
A
B
C
50
60
70
80
90
100
.5
.6
.7
.8
RATIO
REVOLUTION CONSTANT = $C_R = \frac{RD}{V}$ = REVOLUTIONS PER MINUT
SPEED WITH STANDARD WAKE EQUI

Plate 1.

Chart showing the relation between Diameter, Pitch, Helecoidal Area, Indicated Horsepower, Revolutions, Speed and Efficiency. Based on experiments with model propellers by R.E. Froude (Trans. Inst. Naval Architects, XXVII P. 250 and on a discussion of the same in "Marine Propellers" by Sydney W. Barnyby.



Notation.

D = Diameter in feet.

P = Pitch in feet.

$\frac{P}{D}$ = Pitch ratio

$A = \frac{\pi D^2}{4}$ = Disk area

Helicoidal area ÷ A = Disk area ratio

R = Revolutions per minute

$V = M \times S$ = Speed in knots

S = Actual speed of vessel

M = Wake coefficient

A-B Line of increasing values of M where $M \times S = V$

C-D Line of change for varying diameter constant $\frac{H.A.}{D.A.}$

BUREAU OF STEAM ENGINEERING
NAVY DEPARTMENT.
WASHINGTON, D.C. FEB. 14, 1910

forms used, in order to prevent excessive length of hub, are shown in full lines.

For any other ratio of P.A. to D.A. than .32 the new projected area is laid down by increasing or decreasing the circular arc measurements of the .32 area in the ratio of the new area to this .32 standard.

By working on these lines, for any pitch ratio, the resistance arm of the blade always remains the same, no matter what the developed area ratio, and the only change in resistance to turning is that caused by the additional surface. The forms of projection on the disc and on the fore-and-aft plane remain constant for all pitch ratios, and the blades meet fully the following requirement, given by Naval Constructor Taylor and deduced by him from model-tank experiments. "A good practical rule would seem to be to make the blades broader at the tips for low pitch ratios and narrow them for the high ones."

The Bureau blades follow this rule, but in an improved manner, as the broadest part of the developed blade, as measured on the developed elliptical arc, moves slowly in towards the hub as the pitch ratio increases.

The standard form of projected area used by the Bureau of Steam Engineering is what might be called a composite of the very best blades of many that have been investigated, and in passing from the highest to the lowest values of the ratios of P.A. to D.A. agrees very closely with the best blades of these ratios.

By adopting a standard form and value of ratio of the projected to the disk area we reap immediately an additional advantage. In case we wish to use Taylor's formulas for effective and gross horsepower in calculating the efficiency of any proposed propeller we can calculate, and the Bureau has calculated, the values of his coefficients A, B and C for a ratio of P.A. to D.A. of .32, and where any other ratio is to be used the new values of A, B and C are readily obtained by multiplying these calculated values of A, B and C by the ratio between the new P.A. ratio and .32.

Using this standard form of projected area, and the results

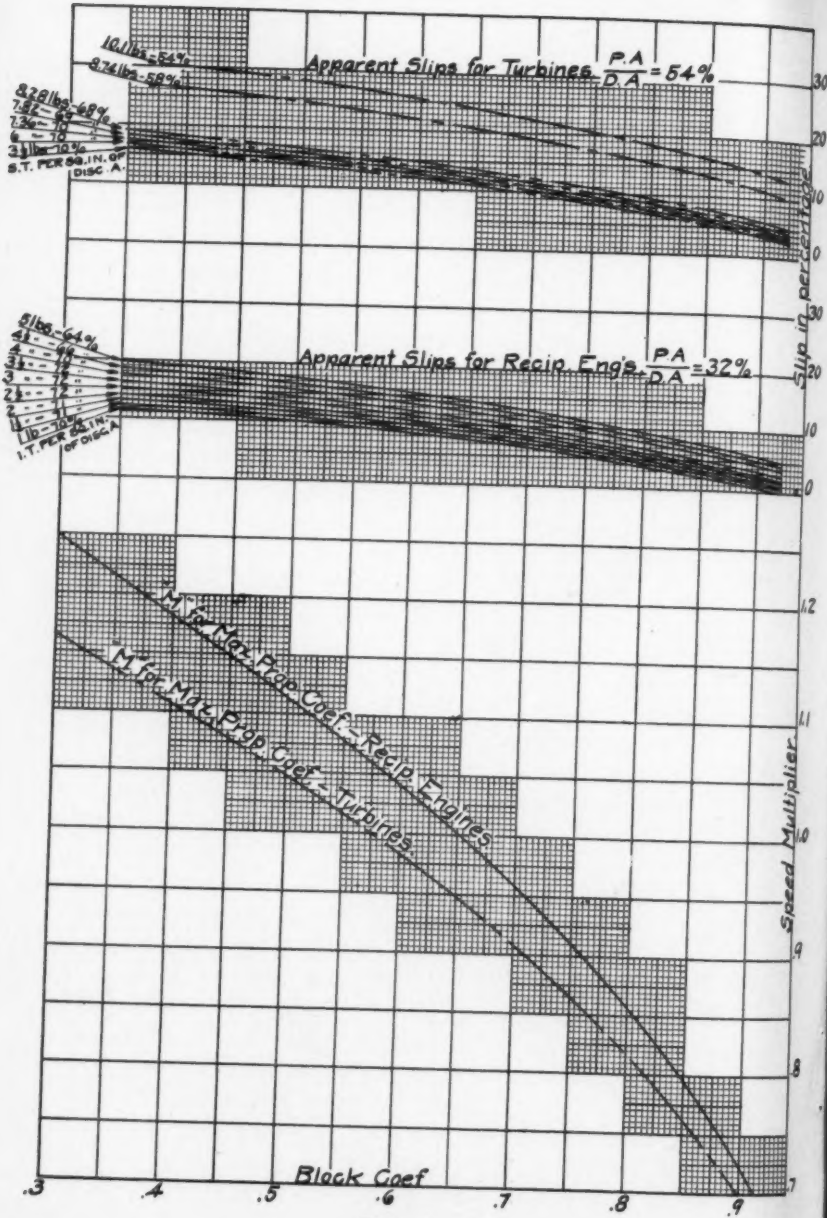

Apparent Slips for Turbines $\frac{P.A}{D.A} = 54\%$
10.1 lbs = 54%
8.74 lbs = 58%
8.28 lbs = 68%
7.82 " = 66 "
7.36 " = 64 "
6 " = 70 "
3½ lbs = 10 %
S.T. PER 50 IN. OF DISC A.
Apparent Slips for Recip. Eng's $\frac{P.A}{D.A} = 32\%$
8 lbs = 64 %
4½ " = 66 "
4 " = 68 "
3½ " = 70 "
3 " = 72 "
2½ " = 72 "
2 " = 71 "
1½ " = 70 %
1 lb = 10 %
I.T. PER 50 IN. OF DISC A.
Slip in percentage
30
20
10
0
30
20
10
0
"M" for Mat. Prop. Coef. - Recip. Engines
"M" for Mat. Prop. Coef. - Turbines
Speed Multiplier
1.2
1.1
1.0
.9
.8
.7
Block Coef
.3
.4
.5
.6
.7
.8
.9

Plate 2.

of many trials over the measured knot, in which great care was taken to insure as accurate results as possible, a chart of design, shown as Plate 5, has been developed, from which a propeller for any particular case can be calculated. The method of development of this chart of design, together with several problems showing how it should be handled, is given in the following.

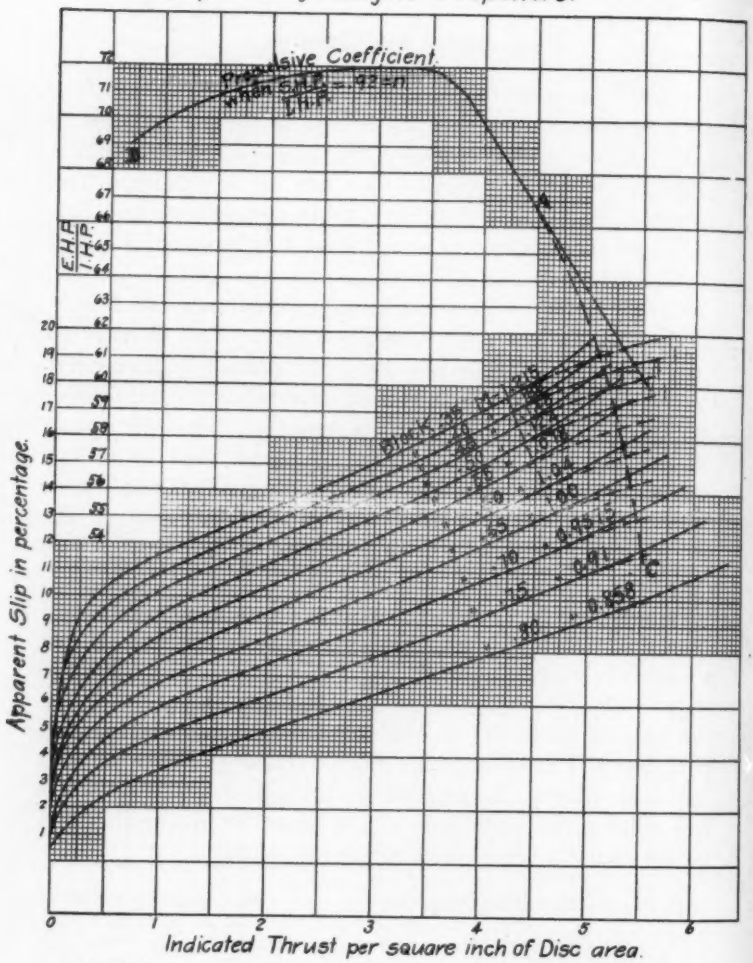
A chart for design has first been prepared, based on Barnaby's application of Froude's results, Plate I. This chart gives the values of C_R for abscissae and C_A for ordinates. This chart has been applied in the reverse manner and, from actual trial results, and for different values of indicated thrust per square inch of disc area, from 1 pound per square inch up to $13\frac{1}{2}$ pounds per square inch, the corresponding values of C_R have been taken from the chart and, from these values of C_R the proper wake factors to use in multiplying the speed, where $V = M \times S$, have been computed.

In applying this chart, if we take the actual speed, revolutions and indicated horsepower developed on trial and assume a value of M equal to unity, we will derive certain values of C_A and C_R. Entering the chart with these values and using the ratio of helicoidal to disc area of the propellers used on trial, we plot a point on the chart. Through the point thus obtained draw a line parallel to the line AB. Where this line crosses the pitch ratio of the propeller in question will be the actual point of plotting for this screw.

Now, in the formula for $C_R = \frac{R \times D}{V}$, we have the value of C_R corresponding to the point as found above, also R and D, which are revolutions and diameter of screw. From this equation we then find the unknown quantity, $V = \frac{R \times D}{C_R}$, and, having the speed of ship over the measured mile, we obtain the value of $M = \frac{V}{S}$.

These values of M for a large number of vessels have been laid down on a chart whose abscissae are block coefficients of fineness with ordinates of M. Having laid down these values

Reciprocating Engine Propellers.



Note:-

I. T per sq.in of Disc area. = $\frac{I.H.P. \times 33000}{P \times R \times A in^2 \times 144}$.

S = Speed in knots.

$C_A = \frac{AV^3}{I.H.P.}$ A = Disc area

$C_R = \frac{R \times D}{V}$ R = Revolutions.

D = Diam. of screw in feet.

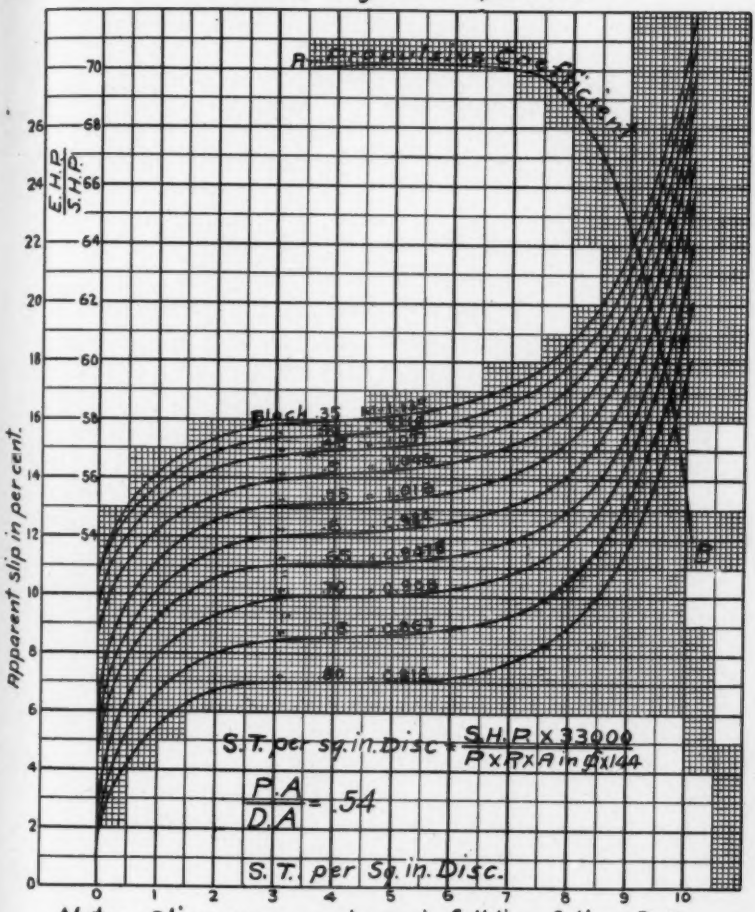
Slips for maximum efficiencies with varying values of Block Coefficient and constant values of "M" for each value of Block Coefficient.

$M = \frac{V}{S}$ for use with Barnaby Chart of Design.

$B - A - \frac{P.A.}{D.A.} = .32$; A - C - I.T. per □" P.A. not to exceed 15 lbs.

Plate 3.

Turbine Engine Propellers.



Note:- Slip curves as shown in full lines follow A-B.

S = Speed in knots.

$C_A = \frac{AV^3 \times .92}{S.H.P.}$ A = Disc Area.

$C_R = \frac{R \times D}{V}$ R = Revolutions. D = Diam. of screw in feet.

Slips for maximum efficiencies with varying values of Block Coefficient and constant values of 'M' for each value of Block Coefficient.

$M = \frac{V}{S}$ for use with Barnaby Chart of Design.

Plate 4.

of M, those vessels with propellers having approximately the same shape of projected area and same ratio of projected to disc area which gave the maximum propulsive coefficients under different conditions of thrust per square inch of disc were selected, and a fair curve run through the corresponding values of M, this curve crossing the zero value of M at a coefficient of fineness value of unity.

The curves of M which remained the more nearly constant through a high range of speeds, and for which the propulsive coefficients averaged the highest, were, for large diameters and high pitch values, those corresponding to a projected area ratio of .32, and for small diameters and low pitches those corresponding to a projected area ratio of .54, the former of these conditions being that which exists in reciprocating-engine practice and the latter at the entrance to the turbine field.

The variation in these curves of M for the different values of thrusts per square inch of disc area for the best blades of similar form (Plate 6) was very small and justified laying down a mean curve of M. This was done for propellers having a projected area ratio to disc area of .32 and for propellers having a similar ratio of .54. Then with these two curves of M, and with abscissae of M and slip percentages (apparent) as ordinates, curves of slip were laid down, and also curves of propulsive coefficients. See Plate 2, where these data were rearranged on abscissae of block coefficients.

These curves of slip and of propulsive coefficients were rearranged on abscissae of indicated thrust per square inch of disc area, and ordinates of slip and propulsive coefficients (Plates 3 and 4 and final chart, Plate 5).

DESCRIPTION OF CHART FOR PROPELLER DESIGN, PLATE 5.

There are shown on Plate 5, first, the curves of propulsive coefficients for value of $\frac{\text{P.A.}}{\text{D.A.}} = .32$ and .54 by the heavy-line curve marked DEF and .32, and by the heavy-line curve marked .54.

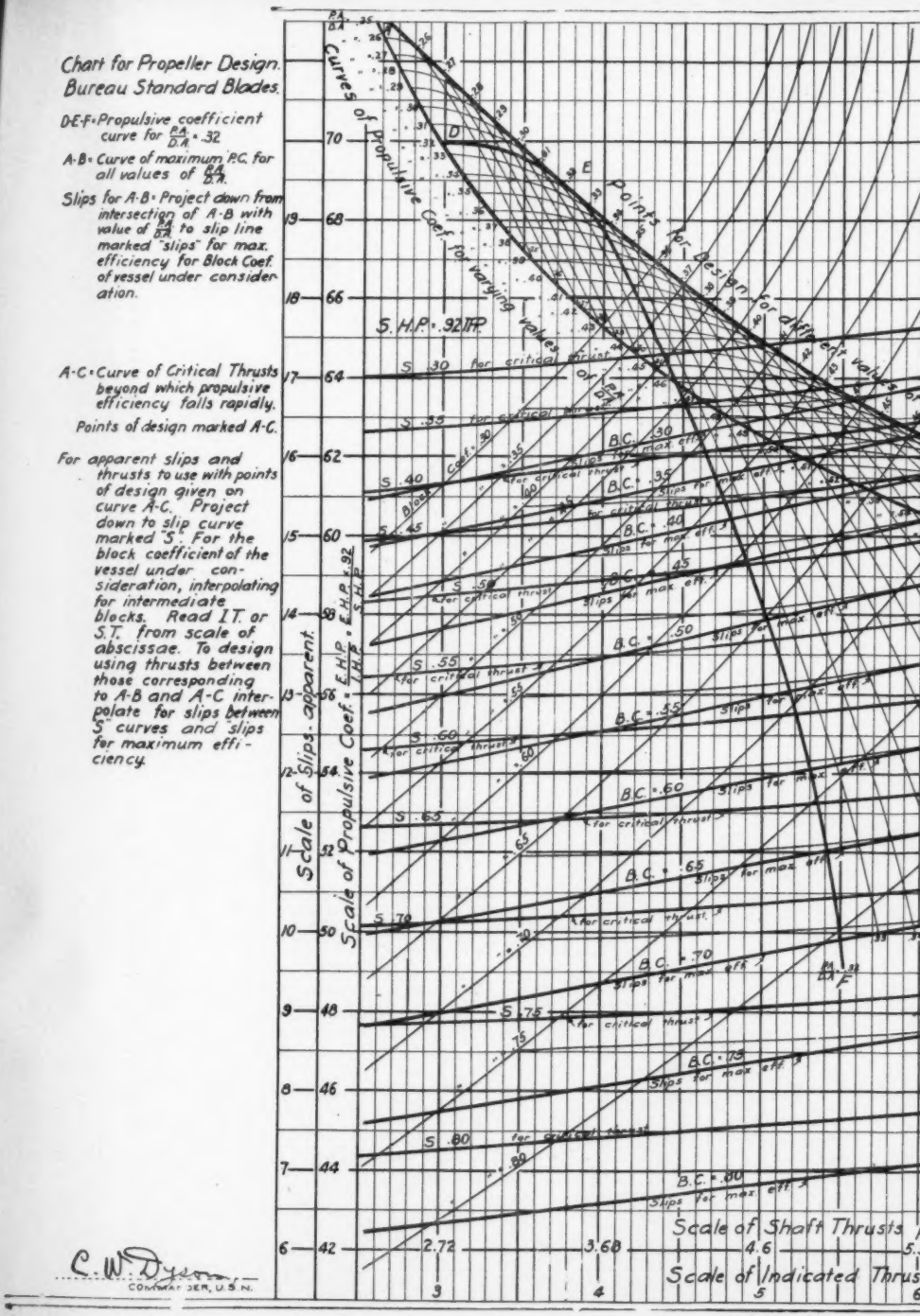


Chart for Propeller Design.
Bureau Standard Blades.
D-E-F = Propulsive coefficient curve for P.A./D.A. = .32
A-B = Curve of maximum P.C. for all values of P.A./D.A.
Slips for A-B: Project down from intersection of A-B with value of P.A./D.A. to slip line marked "slips" for max. efficiency for Block Coef. of vessel under consideration.
A-C = Curve of Critical Thrusts beyond which propulsive efficiency falls rapidly.
Points of design marked A-C.
For apparent slips and thrusts to use with points of design given on curve A-C. Project down to slip curve marked "S". For the block coefficient of the vessel under consideration, interpolating for intermediate blocks. Read I.T. or S.T. from scale of abscissae. To design using thrusts between those corresponding to A-B and A-C interpolate for slips between "S" curves and "slips for maximum efficiency.
Curves of Propulsive Coef. for varying values of P.A./D.A.
Points for design for different values of P.A./D.A.
S.H.P. = .92 I.H.P.
Scale of Slips-apparent.
Scale of Propulsive Coef.: E.H.P./I.H.P. = E.H.P. x .92 / S.H.P.
Scale of Shaft Thrusts
Scale of Indicated Thrus
C. W. Dyson
COMMANDER, U.S.N.

Plate 5.

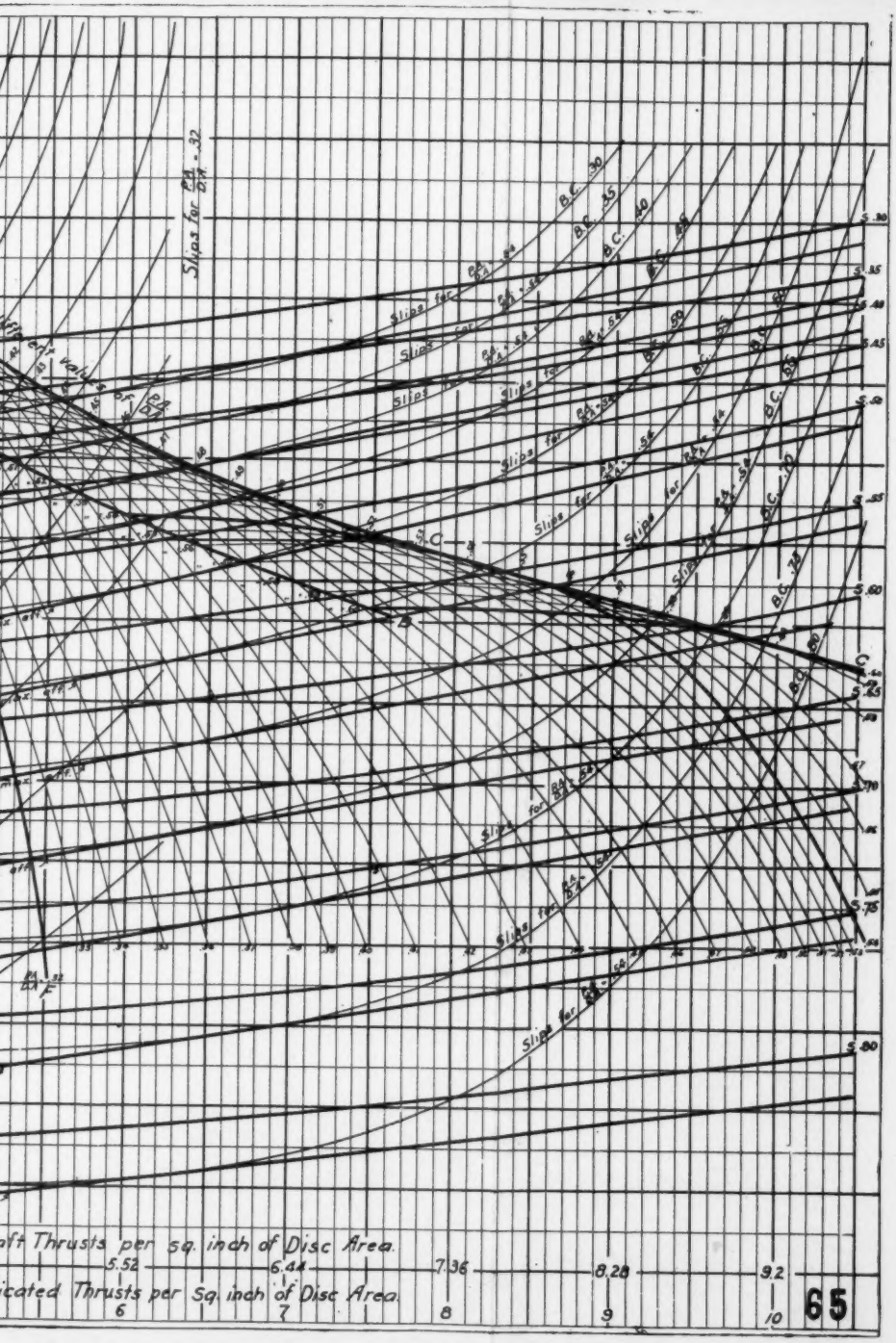

Slips for $\frac{P_s}{D^2N^2}$ = .32
B.C. 30
B.C. 35
B.C. 40
S.C.
C
5.52
6.44
7.36
8.28
9.2
6
7
8
9
10
65

Plate 5.

The slips are shown in light lines marked "Slips for $\frac{P.A.}{D.A.} = .32$" and "Slips for $\frac{P.A.}{D.A.} = .54$." These slips are shown for block coefficients from .3 to .8, which covers nearly the entire range of hulls.

To obtain the propulsive coefficient curves for other values of $\frac{P.A.}{D.A.}$, lines are drawn from points on DEF to points on propulsive coefficient curve .54, where an equal thrust per square inch of projected area occurred, and, assuming that the propulsive coefficient changes proportionately with the change in projected area in passing from the condition of DEF to the condition of .54, these lines are divided each into twenty-two equal parts, and through the corresponding points of division on each line curves are drawn which are shown marked .33, .34, etc. Upon inspection it is found that these curves all have their vertices in a hyperbolic or parabolic curve AB, and have points of critical deflection such that the curve AC will pass tangent to all of these curves of propulsive coefficients, AB and AC approaching each other and probably touching at the point of maximum attainable efficiency.

Dropping down on the chart to the slip lines, the slips corresponding to the line of propulsive coefficients DEF are shown in light curves, marked slips for $\frac{P.A.}{D.A.} = .32$, and are given, as stated before, for blocks from .3 to .8 in fineness, as are also the slips for $\frac{P.A.}{D.A.} = .54$.

Projecting down from the intersection of DEF with AB to each of the slip curves for $\frac{P.A.}{D.A.} = .32$, we find the slips required for maximum possible propulsive coefficient with a propeller whose ratio $\frac{P.A.}{D.A.} = .32$, and doing the same for intersection of propulsive coefficient line for $\frac{P.A.}{D.A.} = .54$ to the

corresponding slip lines, we have the slips for maximum propulsive coefficients with this latter projected area ratio. Now all other points for maximum propulsive coefficients, corresponding to other values of $\frac{\text{P.A.}}{\text{D.A.}}$, will lie on a line passing through these two points, and we have no reason to believe that this line will not be a straight line, or if not a straight line, will be of such slight curvature that, within the ranges of thrust covered, the curvature may be neglected. We therefore assume it a straight line, and through the corresponding points found on the slips for $\frac{\text{P.A.}}{\text{D.A.}} = .32$ and for $\frac{\text{P.A.}}{\text{D.A.}} = .54$, draw the heavy straight lines for each block coefficient of fineness, and mark them for each one as "Slips for Maximum Efficiency."

In the same manner, by projecting down to the slips for $\frac{\text{P.A.}}{\text{D.A.}} = .32$ and $\frac{\text{P.A.}}{\text{D.A.}} = .54$, from the points of tangency of DEF and .54 with curve AC, we find two slips corresponding to the critical thrusts for each block coefficient. By inspection, we see that as the thrusts increase the slip for maximum propulsive coefficient draws closer to the slip for critical thrust, so we pass through each pair of these latter points curves which tend to approach to coincidence with the lines of "Slips for Maximum Efficiency" as the thrusts increase. These curves are marked "S for critical thrust." To obtain slips for maximum propulsive coefficient for any value of $\frac{\text{P.A.}}{\text{D.A.}}$, project down from the intersection of the corresponding curve of propulsive coefficient with AB to the line of "Slip for Maximum Efficiency" for the block coefficient under consideration, which, if not shown, can easily be interpolated for.

In the same manner, by projecting down from the points of tangency of the propulsive coefficient curves with AC to the lines of "Slips for Critical Thrust" or interpolated lines for other blocks, we obtain the slip for critical thrust for any

value of $\frac{\text{P.A.}}{\text{D.A.}}$. Having the slips for maximum efficiency and for critical thrust for any block coefficient and any value of $\frac{\text{P.A.}}{\text{D.A.}}$, we can draw a straight line between these two slip points and project down to it for the slip corresponding to any value of thrust corresponding to any value of propulsive coefficient between AB and points of tangency on AC. Should it be necessary to use a higher thrust than that corresponding to the critical thrust, these lines of slip drawn between the lines of "Slips for Maximum Efficiency" and "Slips for Critical Thrusts" may be extended with a slight curvature upward, the curvature of the slip lines for $\frac{\text{P.A.}}{\text{D.A.}} = .32$ and for $\frac{\text{P.A.}}{\text{D.A.}} = .54$ being used as a guide in determining this amount of curvature, and the slip then measured by projecting down from the point on the curve of propulsive coefficients to this extended slip line.

POINTS TO BE NOTED.

1. Propulsive coefficient $= \frac{\text{E.H.P.}}{\text{I.H.P.}} = \frac{\text{E.H.P.} \times .92}{\text{S.H.P.}}$.

2. E.H.P. = effective horsepower required to tow the ship when the hull is fitted with all appendages.

3. As value of $\frac{\text{P.A.}}{\text{D.A.}}$ decreases, maximum possible propulsive coefficient increases.

4. As value of $\frac{\text{P.A.}}{\text{D.A.}}$ decreases, point of critical thrust approaches more nearly to the point of maximum propulsive coefficient until $\frac{\text{P.A.}}{\text{D.A.}}$ becomes zero, when the curves AB and AC coincide and our blade has become a line.

5. The lower the value of $\frac{\text{P.A.}}{\text{D.A.}}$ the more rapidly does the propulsive coefficient rise to a maximum, reach the critical

point of thrust, and the more rapidly does it fall after passing the point of critical thrust.

6. Converse of 5. The greater the value of $\frac{\text{P.A.}}{\text{D.A.}}$ the less rapidly does the propeller reach its maximum propulsive coefficient, the greater the range of thrust through which it holds near to its maximum before reaching the point of critical thrust, and the slower will be the fall in the propulsive coefficient after passing the point of critical thrust.

7. For best economy at cruising speeds:

(*a*) For vessels to steam constantly at one speed, design by using thrusts corresponding to curve AB.

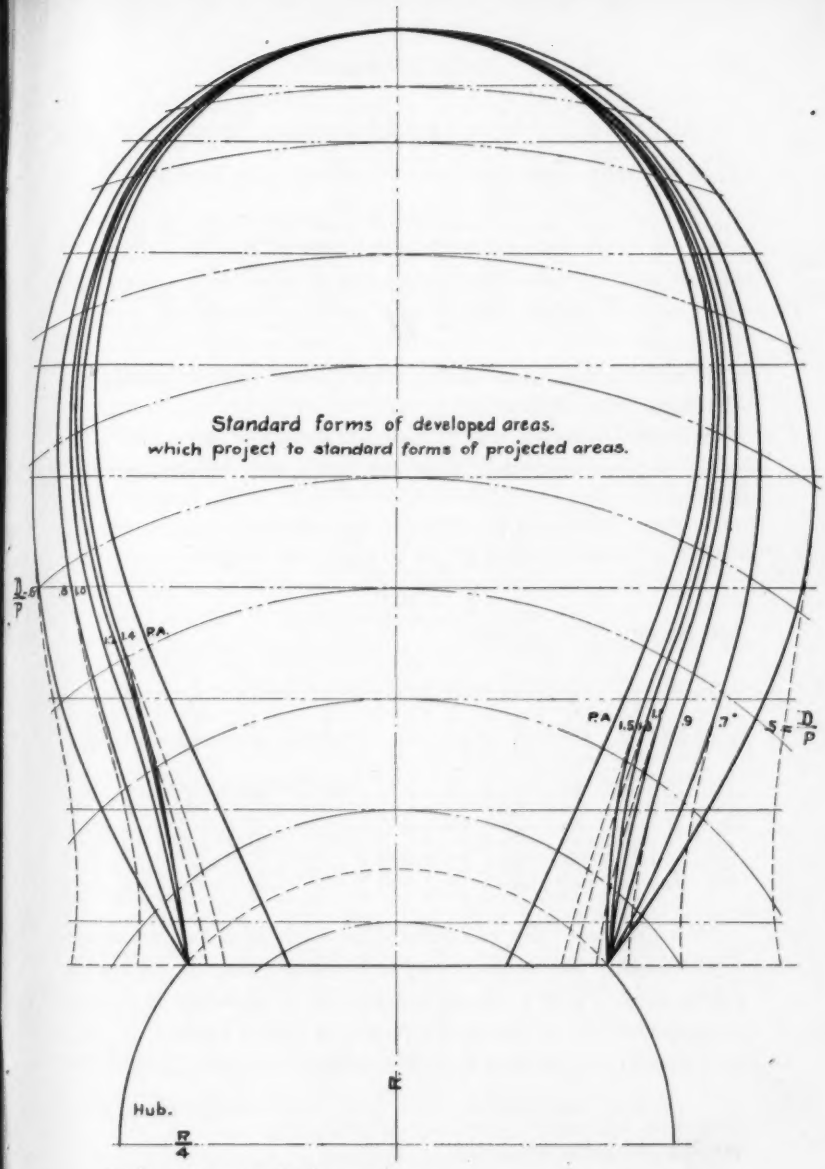
(*b*) For vessels whose cruising speed is slightly below maximum designed speed, design by using thrusts between AB and AC, up to thrusts corresponding to AC.

(*c*) For vessels having reciprocating engines the conditions may be such that a narrow-bladed propeller of sufficient diameter to absorb the power cannot be obtained by designing between AB and the critical thrust. In such case, while a slightly higher speed can be obtained by using a higher value of $\frac{\text{P.A.}}{\text{D.A.}}$, it is obtained at the expense of economy at the lower speeds. It is preferable to use the narrower blades and design for higher thrusts than the critical thrust. This will result in a slightly lower maximum speed than the broader blades would give, but the propulsive coefficients at the lower speeds will be greater.

8. For propellers, to be used with turbine engines in which the ratio $\frac{\text{P.A.}}{\text{D.A.}}$ exceeds .59, we may exceed considerably the critical thrusts given by the point of tangency to AC, as the propulsive efficiency curves for these ratios depart from AC very slowly.

Solution of Problems.

Collier; twin-screw; 95 revolutions per minute; speed, 12 knots; E.H.P. = 2,800. Greatest efficiency of propeller required at speed stated. Block coefficient of fineness = .8.



Projected area = .32 disc area.

Dotted lines are developments of projected areas as shown.

Full " " = actual developments of blades.

For other projected area ratios increase or decrease projected area shown proportionally on circular arcs.

Plate 6.

I.T., per sq. in. D.A.	$\frac{P.A.}{D.A.}$	Propulsive coefficient.	E.H.P., 1 propeller.	H.P.	Slip, per cent.	Revolutions.	S. × 101.33.	P. × R.	Pitch, feet.	Area, square feet.	Diameter, feet.
2.55	.25	.730	1,400	1,918	6.23	95	1,215.96	1,296.75	13.65	135.65	13.2
2.6	.26	.726	1,400	1,929	6.24	95	1,215.95	1,296.89	13.652	131.10	12.95
2.66	.27	.722	1,400	1,940	6.25	95	1,215.96	1,297.02	13.653	128.86	12.85
2.71	.28	.718	1,400	1,950	6.26	95	1,215.96	1,297.16	13.654	127.10	12.75
2.77	.29	.713	1,400	1,964	6.27	95	1,215.96	1,297.30	13.655	125.25	12.65
2.83	.30	.708	1,400	1,978	6.28	95	1,215.96	1,297.44	13.657	123.45	12.55
2.90	.31	.704	1,400	1,989	6.29	95	1,215.96	1,297.58	13.659	121.13	12.45
3.00	.32	.700	1,400	2,000	6.30	95	1,215.96	1,297.71	13.66	117.73	12.25

It will be seen that our necessary total engine power for the speed need not exceed 4,000.

Suppose, however, our engines are designed for 2,500 H.P. each, at 95 revolutions, then our table will be modified as follows:

$$\text{H.P.} : \text{H.P.}_2 : : 95^3 : x^3,$$

$$x = \sqrt[3]{\frac{95^3 \times \text{H.P.}_2}{\text{H.P.}}},$$

and we have H.P.=2,500.

I.T. per sq. in. D.A.	$\frac{P.A.}{D.A.}$	Propulsive coefficient.	E.H.P., 1 propeller.	$H.P._2$	Slip, per cent.	Revolutions $=x$.	S × 101.33.	P × R.	Area, square feet.	Diameter, feet.	Pitch, feet.
2.55	.25	.730	1,400	1,918	6.23	87.00	1,215.96	1,296.75	135.65	13.20	14.90
2.60	.26	.726	1,400	1,929	6.24	87.10	1,215.96	1,296.89	131.10	12.95	14.89
2.66	.27	.722	1,400	1,940	6.25	87.25	1,215.96	1,297 02	128 86	12.85	14 86
2.71	.28	.718	1,400	1,950	6.26	87.40	1,215 96	1,297.16	127.10	12.75	14.84
2.77	.29	.713	1.400	1,964	6 27	87.60	1,215.96	1,297.30	125.25	12.65	14.81
2.83	.30	.708	1,400	1,978	6.28	87 80	1,215.96	1,297.44	123.45	12.55	14.78
2 90	.31	.704	1,400	1,989	6.29	88.00	1,215 96	1,297.58	121.13	12.45	14.75
3.00	.32	.700	1,400	2,000	6.30	88 20	1,215.96	1,297.71	117.73	12.25	14.72

Now suppose the conditions are such that the diameter allowed must not exceed 12.75 feet. According to the above tables, from Table 1, where the engine is working at maximum power for 12 knots, the propeller would have:

Diameter of 12 feet 9 inches; pitch, 13 feet $7\frac{7}{8}$ inches; $\frac{\text{P.A.}}{\text{D.A.}} = .28$; revolutions, 95; H.P., both engines, 3,900; propulsive coefficient, .718.

While according to the second table, in which we have some engine power in reserve, the propeller would have:

Diameter of 12 feet 9 inches; pitch, 14 feet 10 inches; $\frac{\text{P.A.}}{\text{D.A.}} = .28$; revolutions, 87.4; H.P., both engines, at 12 knots, 3,900; propulsive coefficient, .718.

In naval vessels, where the cruising speeds are much lower than the maximum speeds, we proceed in the same manner as in the above problem, except we use the thrusts corresponding to the points of tangency on the curve AC instead of those corresponding to the intersection points on AB. In these cases, however, we are usually supplied with a curve of E.H.P., which extends considerably beyond the contract speed, so that it is possible to use the full designed power of the engine or such maximum power as it may have developed on previous trials, and, with the propulsive coefficient corresponding to the P.A. of the propeller being considered, and this E.H.P. curve, obtain the possible speed with this power and use it in the calculations, thus:

H.P., two engines, = 20,000; revolutions, 125 per min. Use $\frac{\text{P.A.}}{\text{D.A.}} = .32$. Diameter allowed, about 18 feet. Block coefficient of fineness of vessel = .66.

H.P.	I.T., per sq. in. D.A.	Prop. coef.	E.H.P.	Speed from curve.	Revolutions.	Slip, per cent.	Pitch.	Diameter, feet.	P.A., square feet.
10,000	3	.72	7,200	19.36	125	10.0	17.44	21.12	114.7
10,000	3½	.718	7,180	19.35	125	10.8	17.58	19.47	95.32
10,000	4	.698	6,980	19.26	125	11.6	17.66	18.18	83.04
10,000	4½	.682	6,820	19.20	125	12.2	17.73	17.6	77.87
10,000	4¾	.666	6,665	19.14	125	12.6	17.75	17.1	73.44

Make propeller 18 feet 2 inches diameter; pitch, 17 feet 8 inches; P.A., 83.04 square feet.

Let us now take the case of a vessel propelled by turbine engines, and take as our preliminary data the following:

Total S.H.P. = 12,000; number of shafts, 3, power equally divided between the three shafts; revolutions, 800 per min-

ute; contract speed, 29½ knots. Block coefficient of fineness of vessel = .414.

Equivalent I.H.P. on one shaft = $\frac{4,000}{.92}$ = 4,348;

From E.H.P. curve, E.H.P. for contract speed = 6,100;
E.H.P. on one shaft = 2,033.

$\frac{P.A.}{D.A.}$	Prop. coef.	Speed.	E.H.P.	I.H.P.$_1$	Slip.	Revolutions.$_1$	101.33×29.5.	P.×R.	P., feet.	I.T., per sq. in. D.A.	Disk.	Diameter, feet.	$\frac{P.}{D.}$
.58	.582	29.5	2,033	3,493	17.2	744	2,989.24	3,610.19	4.852	9.31	23.82	5.5	.882
.59	.578	29.5	2,033	3,517	17.4	745	2,989.24	3,618.93	4.857	9.65	23.13	5.4	.899
.60	.575	29.5	2,033	3,536	17.6	747	2,989.24	3,627.72	4.856	10.00	22.34	5.34	.909
.61	.572	29.5	2,033	3,555	17.8	748	2,989.24	3,636.54	4.862	10.36	21.63	5.25	.926

$$\text{Revs.}_1^3 : \text{Revs.}_2^3 :: \text{H.P.}_1 : \text{H.P.}_2,$$

$$\text{Rev.}_1 = \sqrt[3]{\frac{\text{Rev.}_2^3 \times \text{H.P.}_1}{\text{H.P.}_2}} = \sqrt[3]{\frac{800 \times \text{H.P.}_1}{4,348}}.$$

Any of the above propellers will be satisfactory, and we should be governed in our choice by diameter limitations.

SUPPLEMENTARY PROPELLERS.

For any propeller of any value of $\frac{P.A.}{D.A.}$ there is a supplementary propeller of a different value of $\frac{P.A.}{D.A.}$, and different pitch and diameter for every value of thrust per square inch of disc area, which will give the same value of propulsive coefficient at this one point of thrust as the original propeller.

To show this point more thoroughly the following problem has been solved, the results of which are shown in the accompanying table and chart of faired curves (Plate 7).

Block coefficient of vessel = .60.
E.H.P. (total with all appendages) = 10,000.
E.H.P. for one propeller = 5,000.
Revolutions = 125 per minute.
Speed = 18 knots.

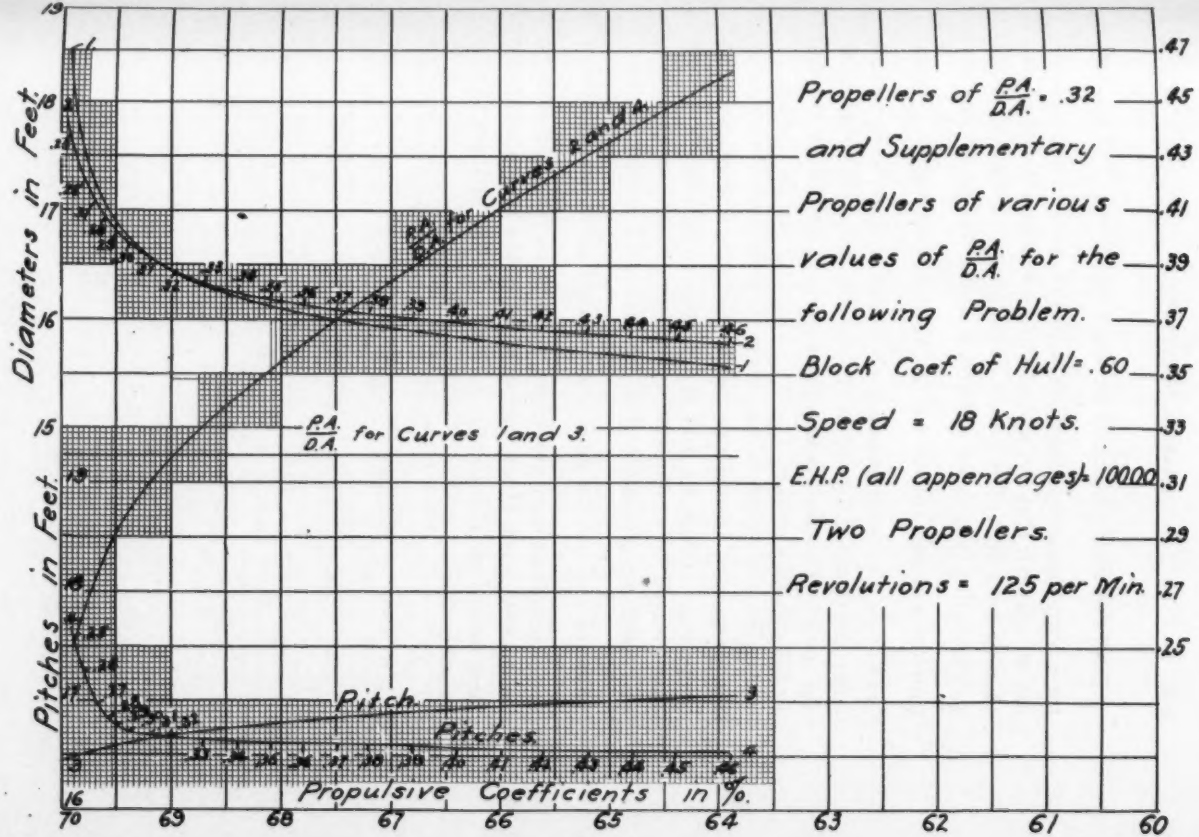

Diameters in Feet.
Pitches in Feet.
Propulsive Coefficients in %.
P.A. / D.A. for Curves 1 and 3.
P.A. / D.A. for Curves 2 and 4.
Pitch.
Pitches.
Propellers of P.A. / D.A. = .32 and Supplementary Propellers of various values of P.A. / D.A. for the following Problem.
Block Coef. of Hull = .60
Speed = 18 Knots.
E.H.P. (all appendages) = 10000.
Two Propellers.
Revolutions = 125 per Min.

Plate 7.

Find a series of propellers having a value of $\frac{P.A.}{D.A.}$ of .32, and their supplementary propellers with this value varying from .25 to .46, such that the propulsive coefficient will not be less than 64 per cent. The points of design are shown on Plate 5, where the propulsive coefficient curve for .32 crosses the propulsive coefficient curves for various values of $\frac{P.A.}{D.A.}$ from .25 to .46, the corresponding slips and thrusts being taken from the chart and entered in the slip and I.T. columns of the table. The resultant pitches and diameters are entered in the columns marked $P_{.32}$, $P_{Sup.}$, $D_{.32}$, $D_{Sup.}$, in the Table.

Attention is called to the slow and very gradual change in the propulsive coefficients and in the characteristics of the propellers. When we look at the great number of propellers here arrayed for a comparatively small change in efficiency, and when we remember that we can make screws for each value of $\frac{P.A.}{D.A.}$ for the same work, each of these having its own set of supplementary propellers, it is readily seen how vast the field for criticism of any particular propeller becomes and also how hazardous it is to offer adverse criticism without careful investigation.

PROPELLERS OF $\frac{\text{P.A.}}{\text{D.A.}} = .32$ AND THEIR SUPPLEMENTS.

BLOCK COEF., .60.

P.A. / D.A.	I.T.	E.H.P.	Prop. coef.	Slip.	S.	I.H.P.	Revolutions.	S. × 101.33.	P. × R.	$P_{.32}$, feet.	$P_{sup.}$, feet.	$Area_{.32}$, square feet.	$Area_{sup.}$, square feet.	$D_{.32}$, feet.	$D_{sup.}$, feet.
.32 ...	3.18	5,000	69.9	11.55	18	7,153	125	1,823.94	2,062.12	16.497	...	260.7	...	18.25	...
... .25	3.18	5,000	69.9	17.00	18	7,153	125	1,823.94	2,197.58	...	17.580	...	244.6	...	17.7
.32 ...	3.26	5,000	69.8	11.65	18	7,163	125	1,823.94	2,064.45	16.515	...	243.9	...	17.625	...
... .26	3.26	5,000	69.8	15.50	18	7,163	125	1,823.94	2,158.51	...	17.268	...	233.3	...	17.25
.32 ...	3.34	5,000	69.7	11.75	18	7,174	125	1,823.94	2,066.79	16.534	...	238.2	...	17.5	...
... .27	3.34	5,000	69.7	14.00	18	7,174	125	1,823.94	2,120.86	...	16.969	...	232.1	...	17.2
.32 ...	3.43	5,000	69.6	11.90	18	7,184	125	1,823.94	2070.306	16.562	...	231.8	...	17.15	...
... .28	3.43	5,000	69.6	13.30	18	7,184	125	1,823.94	2,103.74	...	16.830	...	228.2	...	17.05
.32 ...	3.53	5,000	69.5	12.10	18	7,194	125	1,823.94	2,075.02	16.600	...	225.07	...	16.9	...
... .29	3.53	5,000	69.5	13.10	18	7,194	125	1,823.94	2,098.89	...	16.791	...	222.5	...	16.85
.32 ...	3.60	5,000	69.4	12.25	18	7,205	125	1,823.94	2,078.56	16 628	...	220.7	...	16 75	...
... .30	3.60	5,000	69.4	12.80	18	7,205	125	1,823.94	2,091.67	...	16.733	...	219.3	...	16.72
.32 ...	3.70	5,000	69.0	12.45	18	7,246	125	1,823.94	2,083.31	16.666	...	215.4	...	16.6	...
... .31	3.70	5,000	69.0	12.70	18	7,246	125	1,823.94	2,089.28	...	16.71	...	214.8	...	16.58
.32 ...	3.80	5,000	68.7	12.65	18	7,278	125	1,823.94	2,088 08	16.705	...	210.2	...	16.30	...
... .33	3.80	5,000	68.7	12.45	18	7.278	125	1,823.94	2,083.31	...	16 666	...	210.7	...	16.375
.32 ...	3 88	5,000	68.4	12.70	18	7,310	125	1,823.94	2,089.28	16.714	...	206.6	...	16.20	...
... .34	3.88	5,000	68.4	12.30	18	7,310	125	1,223.94	2,079.75	...	16.638	...	207.6	...	16.25
.32 ...	3.93	5,000	68.1	12.80	18	7,342	125	1,823.94	2,091.67	16.733	...	204 7	...	16.125	...
... .35	3.93	5,000	68.1	12.20	18	7,342	125	1.823.94	2,077 38	...	16.619	...	206.1	...	16.20
.32 ...	3.98	5,000	67.8	12.90	18	7,374	125	1,823.94	2,094.08	16.752	...	202.8	...	16.07	...
... .36	3.98	5,000	67.8	12.10	18	7,374	125	1,823.94	2,075.02	...	16 600	...	204.6	...	16.125
.32 ...	4.03	5,000	67.5	13.05	18	7,407	125	1,823.94	2,097.69	16.781	...	200.8	...	15.98	...
... .37	4.03	5,000	67.5	12.05	18	7,407	125	1,823.94	2,073.84	...	16.590	...	203.1	...	16.08
.32 ...	4.07	5,000	67.2	13.15	18	7,440	125	1,823.94	2,100.22	16.800	...	199 5	...	15 93	...
... .38	4.07	5,000	67.2	12.02	18	7,440	125	1,823.94	2,073.13	...	16.585	...	202.1	...	16.06
.32 ...	4.11	5,000	66.8	13.22	18	7,485	125	1,823.94	2,101.80	16.814	...	198.6	...	15 90	...
... .39	4.11	5,000	66.8	12.00	18	7.485	125	1,823.94	2,072.66	...	16.580	...	201.4	...	16.02
.32 ...	4.15	5,000	66.4	13.30	18	7,530	125	1,823 94	2,103.74	16.830	...	197.7	...	15.85	...
... .40	4.15	5,000	66.4	11.95	18	7,530	125	1,823.94	2,071.48	...	16.572	...	200.7	...	15.97
.32 .	4.19	5,000	66.0	13.40	18	7 576	125	1,823.94	2,106.17	16.850	...	196 7	...	15.82	...
... .41	4.19	5,000	66.0	11.85	18	7,576	125	1,823.94	2,069.13	...	16.553	...	200.2	...	15.96
.32 ...	4.23	5,000	65.6	14.00	18	7,622	125	1,823.94	2,120.86	16.969	...	194.7	...	15.73	...
... .42	4.23	5,000	65.6	11.78	18	7,622	125	1,823.94	2,067.49	...	16.540	...	199.7	...	15.95
.32 ...	4.28	5,000	65.2	14.10	18	7,669	125	1,823.94	2,123.33	16.986	...	193.5	...	15.66	...
... .43	4.28	5,000	65.2	11.75	18	7,669	125	1,823.94	2,066.79	...	16.534	...	198.7	...	15.94
.32 ...	4.33	5,000	64.8	14.20	18	7,716	125	1,823.94	2,125.80	17.006	...	192.1	...	15.63	...
... .44	4.33	5,000	64.8	11.73	18	7,716	125	1,823.94	2,066.32	...	16.530	...	197.6	...	15.84
.32 ...	4.39	5,000	64.4	14.30	18	7,764	125	1,823.94	2,128.28	17.026	...	190.4	...	15.60	...
... .45	4.39	5,000	64.4	11.71	18	7,764	125	1,823.93	2,065.85	...	16.526	...	196.2	...	15.81
.32 ...	4.44	5,000	63.9	14.40	18	7,825	125	1,823.94	2,130.77	17.046	...	189.5	...	15.57	...
.. .46	4.44	5,000	63.9	11.70	18	7,825	125	1,823.94	2,065.62	...	16.525	...	195.6	..	15.78

Ratio Between Effective Horsepowers Required to Tow Hull of Vessel Without (E.H.P.), and with ($E.H.P._1$) Appendages.

Appendages (3d column): 1. Rudder; 2. Shafts; 3. Struts; 4. Bilge keels; 5. Docking keels; 6. Single keel; 7. Bosses; 8. Swells; 9. Armor.

Vessel.	$\frac{EHP._1}{EHP.}$	Appendages, model towed with.	Type of vessel.	Len'th on L.W.L., feet.	Dis-placement, tons.	Bl'ck coef. of fine-ness.
			2-shaft			
Perry	1.17	1,2,3,4,8	Destroyer.	245	480	.410
Paul Jones	1.17	1,2,3,4,8	"	245	450	.410
Preble	1.17	1,2,3,4 8	"	245	435	.410
Truxtun	1.17	1,2,3,4,8	"	248.5	490.3	.370
Bainbridge	1.17	1,2,3,4,8	"	240.5	460	.430
Lawrence	1.17	1,2,3,4,8	"	240.8	412	.404
			2-shaft			
Thornton	1.095	1,2,3,8	Torpedo boat.	175	174	.383
Craven	1.095	1,2,3,8	"	147	133	.448
Barney	1.095	1,2,3,8	"	157	168	.478
			3-shaft			
Smith	1.090	1,2,3,4,7	Destroyer.	289	700	.430
Lamson	1.090	1,2,3,4,7	"	289	700	.430
Preston	1.090	1,2,3,4,7	"	289	700	.430
Flusser	1.090	1,2,3,4,7	"	289	700	.430
Reid	1.090	1,2,3,4,7	"	289	700	.430
Chester	1.058	1,2,3,4	4-shaft scout.	420	3,977	.407
Salem	1.1134	1,2,3,4	2-shaft scout.	420	4,000	.407
Birmingham	1.1150	1,2,3,4	"	420	3,750	.405
			2-shaft			
Charleston	1.137	1,3,3,4,5	Prot. cruiser.	424	9,710	.539
St. Louis	1.128	1,2,3,4,5	"	424	9,665	.539
Milwaukee	1.116	1,2,3,4,5	"	424	9,700	.539
			2-shaft			
New York	1.176	1,2,3,4,8,9	Arm. cruiser.	380.5	8,480	.509
Brooklyn	1.220	1,2,3,4,8	"	400.5	8,150	520
Maryland	1.170	1,2,3,4,6,8	"	502	13,720	.558
West Virginia	1.170	1,2,3,4,6,8	"	502	13,720	.558
Colorado	1.127	1,2,3,4,6,8	"	502	13,670	.558
Pennsylvania	1.127	1,2,3,4,6,8	"	502	13,670	.558
California	1.170	1,2,3,4,6,8	"	502	13,750	.558
South Dakota	1.170	1,2,3,4,6,8	"	502	13,750	.558
Washington	1.213	1,2,3,4,5	"	502	14,495	.554
Tennessee	1.213	1,2,3,4,5	"	502	14,480	.554
Montana	1.213	1,2,3,4,5	"	502	14,518	.554
North Carolina	1.213	1,2,3,4.5	"	502	14,518	.554
			2-shaft			
Massachusetts	1.153	1,2,3,4	Battleship.	348	10,555	.622
Oregon	1.153	1,2,3,4	"	348	10,250	.620
Iowa	1.280	1,2,3,4	"	360	11,363	.630
Wisconsin	1.246	1,2,3,4,5,8	"	368	11,500	.642
Alabama	1.246	1,2,3,4,5,8	"	368	11,734	.643
Kearsarge	1.246	1,2,3,4,5,8	"	368	11,738	.644
Maine	1.093	1,2,3,4,5,8	"	388	12,755	.654
Missouri	1.093	1,2,3,4,5,8	"	388	12,240	.654
Ohio	1.093	1,2,3,4,5,8	"	388	12,480	.654
Mississippi	1.245	1,2,3,4,5,8	"	375	13,040	.660

RATIOS—*Continued.*

Vessel.	$\frac{EHP_1}{EHP}$	Appendages, model towed with.	Type of vessel.	Len'th on LWL., feet.	Displacement, tons.	Bl'ck coef. of fineness.
Idaho	1.245	1,2,3,4,5,8	2-shaft Battleship	375	13,040	.660
Virginia	1.166	1,2,3,4,5,8	"	435	14,978	.658
Georgia	1.148	1,2,3,4,5,8	"	435	14,907	.656
Nebraska	1.166	1,2,3,4,5,8	"	435	14,900	.656
Rhode Island	1.187	1,2,3,4,5,8	"	435	14,860	.655
New Jersey	1.187	1,2,3,4,5,8	"	435	14,870	.655
Connecticut	1.206	1,2,3,4,5,8	"	450	16,377	.660
Kansas	1.216	1,2,3,4,5,8	"	450	16,075	.660
Louisiana	1.216	1,2,3,4,5,8	"	450	16,033	.660
Minnesota	1.206	1,2,3,4,5,8	"	450	16,016	.660
Vermont	1.206	1,2,3,4,5,8	"	450	15,986	.660
New Hampshire	1.206	1,2,3,4,5,8	"	450	16,023	.660
South Carolina	1.224	1,2,3,4,5,8	"	450	16,000	.640
Michigan	1.224	1,2,3,4,5,8	"	450	16,000	.640
Delaware	1.174	1,2,3,4,5,7,8,9	"	510	20,000	.600
North Dakota	1.174	1,2,3,4,5,7,8,9	"	510	20,000	.600
Mars	1.124	Lundborg stern bilge keels.	Collier.	375	11,200	.799

Column 2 gives the appendages attached to models during towing experiments, from which ratio in column 1 is calculated. The ordinary appendages in service are for different classes of vessels, as follows:

Battleships, 1, 2, 3, 4, 5, 7, 8 and sometimes 9.

Armored cruisers, 1, 2, 3, 4, 5, 7, 8 and sometimes 9.

Heavy protected cruisers, 1, 2, 3, 4, 5, 7, 8.

Light protected cruisers, 1, 2, 3, 4, 7, 8.

Scouts, 1, 2, 3, 4, 7, 8.

Destroyers, 1, 2, 3, 4, 7, 8, and, sometimes, scoops.

Torpedo boats, 1, 2, 3, 4, 7, 8, and scoops.

It is axiomatic that in the case of vessels of differing degrees of fineness, but having equal appendage resistances, the percentage increase of total resistance due to the appendages will be the greatest for that vessel having the least resistance due to bare hull.

The writer is indebted to Naval Constructor D. W. TAYLOR, U. S. N., for the following note:

"It is quite impossible to give definite percentages of total appendage resistance as divided among the various appendages, since the area and resistance of appendages vary so. We generally assume that such appendages as bilge keels, docking keels and rudder offer but little, if any, more resistance than

that due to their surface. The same applies to the bar keel. Bosses and swells are usually included with shafts and struts. The armor is generally made smooth, so that there is no special resistance from that as an appendage. Scoops have been tried only on some destroyer models, where their resistance was about 8 per cent. of the bare hull resistance.

"Any assignment of the total appendage resistance between the appendages named would be practically guess-work, but it is presumed that a reasonable distribution would be as follows:

"Bilge keels and docking keels about 30 per cent. each.

"Struts and shafts about a third, and rudder about $6\frac{2}{3}$ per cent."

ANALYSIS OF A PROPOSED DESIGN.

It often becomes necessary to analyze the probable performance of a propeller that has been proposed for a vessel. In this case we proceed as follows:

Data.

Block coefficient of fineness of vessel = .62.

Proposed propeller, pitch = 102 inches = 8.5 feet.

diameter = 110 inches = 9.167 feet.

Projected area ÷ disc area = .56.

Required to find the revolutions, speed and slip when the main turbines are developing a total of 28,000 S.H.P. on four shafts.

Model-tank curves for the vessel give the E.H.P. required at different speeds, the hull being fitted with all appendages.

$\frac{P.A.}{D.A.}$	Propulsive coefficient.*	E.H.P.†	I.H.P.	V = speed.†	Slip.*	Pitch, feet.	Speed $\times \frac{101.33}{\text{Pitch}}$	$P \times R = \frac{V \times 101.33}{100 - \text{slip.}}$	$R = \frac{P \times R}{P}$.	I.T. per sq. inch disc.*	Disc area, square feet.	Diameter, feet.
.56	58.9	3,475	5,900	20.1	13.12	8.5	2,037.73	2,345.45	275.93	8.7	66.27	9.18
.56	58.9	3,625	6,155	20.3	13.12	8.5	2,057.0	2,367.64	278.54	8.7	68.47	9.35
.56	58.9	3,800	6,452	20 5	13.12	8.5	2,077.27	2,390.96	283.64	8.7	71.08	9.53
.56	58.9	4,200	7,131	20.9	13.12	8.5	2,117.80	2,437.61	286.78	8.7	77.06	9.95

* Data from the propeller chart of design.
† Data from model tank curve.

Curves of I.H.P., revolutions and diameter can now be laid down on abscissae of speed. For any particular speed the ordinate will cut the curves giving the corresponding I.H.P., revolutions and diameter. At the given diameter, 110 inches, a perpendicular is erected and gives us the corresponding I.H.P., revolutions and speed for the propeller under consideration.

Our propeller, we find, corresponds closely to the following: $\frac{\text{P.A.}}{\text{D.A.}} = .56$; E.H.P. $= 3{,}475$; I.H.P. $= 5{,}900$; speed $= 20.1$ knots; revolutions $= 275.93$; diameter $= 9.18$ feet.

But the machinery is designed to give 28,000 S.H.P. $= \frac{28{,}000}{.92}$ I.H.P. on four shafts, or $= \frac{7{,}000}{.92}$ I.H.P. on one shaft.

Now, $\frac{7{,}000}{.92} : 5{,}900 :: R^3 : \overline{275.93^3}$;

$$\text{Revolutions at contract power} = \sqrt[3]{\frac{7{,}000}{.92 \times 5{,}900} \times \overline{275.93^3}}$$

$$= 300.3;$$

$$P \times R = 300.3 \times 8.5;$$

$$\text{I.H.P.} \times 33{,}000 = \frac{7{,}000 \times 33{,}000}{.92};$$

$$\text{Total I.T.} = \frac{7{,}000 \times 33{,}000}{.92 \times 300.3 \times 8.5};$$

Total disc area of propeller $= 9{,}503.32$ square inches;

$$\text{I.T. per square inch disc area} = \frac{7{,}000 \times 33{,}000}{.92 \times 300.3 \times 8.5 \times 9{,}503.32}$$

$$= 10.35.$$

The propulsive coefficient for a P.A. ratio of .56, with an indicated thrust of 10.35 pounds per square inch of disc area, is, from the chart, 54.1 per cent.

Therefore, total E.H.P. for four shafts at 28,000 S.H.P. $= \frac{28{,}000}{.92} \times 54.1 = 16{,}465$.

The speed corresponding to this, from the model-tank curve, $= 20.81$ knots.

Slip $= 17.2$ per cent.

NOTE.—Reprints of this article, in flexible cloth covers, with Plates 1 and 5 on linen, will be supplied by the Secretary-Treasurer of the Society for fifty cents each, postpaid.

COMPARATIVE TRIALS OF SCOUT CRUISERS *BIRMINGHAM, SALEM, CHESTER.*

By Commander W. W. White, U. S. N., Member.

With a view to comparing steam expenditures of reciprocating and turbine-propelling machinery, an extensive series of tests was carried out some months ago on the Scout Cruisers *Birmingham, Salem* and *Chester,* under a Board appointed by the Navy Department. The vessels named were selected for the tests as the hulls are practically of the same model and dimensions, while the propelling machinery differs in all three ships. The *Birmingham* is propelled by reciprocating engines; the *Salem* by Curtis turbines, and the *Chester* by turbines of the Parsons type.

A special water-measuring apparatus was built and installed on each ship for the tests, and the arrangement was such that the steam required, at various speeds, could be, after condensation, accurately ascertained both for the propelling and auxiliary machinery. To make the comparison complete, six distinct sets of tests were made, which may be briefly summarized as follows:

I. *Standardization trials.*—Each vessel was standardized over the Rockland measured-mile course just prior to undertaking (II) steam-consumption trials. This involved standardizing the *Salem* on two different occasions, as two series of steam-consumption tests were carried out. On these trials, and on all other (III and IV) sea runs as well, the aim was to load each vessel so that the average displacement would be 4,000 tons.

II. *Steam-consumption tests of main propelling machinery.*—In these tests the weight of exhaust steam entering the main condensers, at different speeds of the vessels, from about 10 knots up to maximum, was found by means of main meas-

uring apparatus. Incidentally, exhaust steam from all auxiliaries in operation on these trials, as a whole, was ascertained, and it will be apparent that the total boiler output in steam, during any test, amounted to the sum of the weights of water collected in the main and auxiliary tanks. No attempt was made to weigh or otherwise determine the amount of coal used on these trials.

III. *Coal-consumption trials,* consisting of sea runs of about 1,000, 750 and 2,000 knots at speeds, respectively, of 10, 15 and 20 knots per hour. Water-measuring apparatus was removed before commencement of these trials, and attention concentrated on securing weight of coal consumed. From data recorded, however, using results of previous trials, the steam consumption of all machinery on these and the (IV) full-power trial has been estimated.

IV. *Full power run of twenty-fours duration.*—As will appear later, the *Birmingham* completed only twelve hours of this trial because of a developed defect in the propelling machinery.

V. *Boiler evaporative tests.*—Vessels in port.

VI. *Individual tests of auxiliaries.*—Vessels in port.

In setting forth details of the various tests, embracing a part only of the Board's report, the sequence above outlined is followed, although the trials were not carried out in that order.

DESCRIPTION OF VESSELS.

A full description of the scout cruisers, in connection with data of official contract-acceptance trials, has appeared in previous numbers of the JOURNAL (Volume XX, pages 349, 706 and 976), to which the reader is referred for extended information covering hull and machinery details of each ship. The important particulars are set forth below:

PRINCIPAL HULL DATA.

The hulls (plate 1) of all vessels, except in minor details, are alike. Steel is used throughout, and the outside plating generally is on the raised and sunken system. The frames,

spaced 36 inches apart, are generally of channel section, 6 by $2\frac{13}{16}$ by $2\frac{13}{16}$ by 13.3 pounds.

The outside plating, below the load water line, consists of 15-pound plating, reduced to 12-pound at the ends, and of 12-pound plating elsewhere. Flat keel plates, about 36 inches in width, are in two thicknesses; the inner of $17\frac{1}{2}$ and the outer of 20-pound plate. The vertical keel, about 39 inches in depth, is of 15-pound plate. Garboard and sheer strakes are $17\frac{1}{2}$ pounds, reduced to 15 pounds at the ends. Nickel-steel protection (80-pound plate) of variable width, extends fore and aft in wake of engine, boiler and dynamo rooms.

The weight of hull, including nickel-steel protection, but without machinery, coal, stores, outfit, armament and ammunition, etc., is approximately 2,015 tons. Principal dimensions are:

Length between perpendiculars, feet	420
Length over all, feet and inches	423—2
Length on L.W.L., feet	420
Breadth, molded, feet and inches	46—8
Breadth, extreme, feet and inches	47—0¾
Ratio of length to beam	8.97
Draught (official contract trial, 3,750 tons displacement), feet and inches	16—9
4,000 tons displacement, feet and inches	17—4½
4,710 tons displacement (about fully loaded), feet and inches	19—2¾
Displacement per inch at mean draught (16 feet 9 inches), tons	31.07
Area of midship section (3,750 tons displacement), square feet	566
Area of L.W.L. section (3,750 tons displacement), square feet.	12,960
Wetted surface section (3,750 tons displacement), square feet.	19,900
Coefficient (at 3,750 tons displacement):	
Block	.40
Midship	.72
L.W.L. plane	.66

MACHINERY INSTALLATIONS.

(Plate 2)

Main propelling machinery, of 16,000 I.H.P., at maximum power, of the twin-screw, reciprocating type, for the three scout cruisers, was designed by the Department. Proposals, however, for the construction of these vessels were invited

under two classes, viz: first, for hull and machinery in accordance with Department's plans and specifications; second, in general accordance with the Department's plans and specifications, but on bidder's design of machinery, preference to be given, other things being equal, to a turbine installation.

After examination of the bids submitted, contracts were awarded on the Department's design of hull, as follows: To the Fore River Ship and Engine Building Company, Quincy, Mass., two vessels—the *Birmingham* and *Salem*—at a contract price of $1,556,000 each; the first named to have the Department's design for propelling machinery, and the latter to be equipped with Curtis marine turbines. Contract for the remaining vessel, the *Chester,* was awarded to the Bath Iron Works, Bath, Me., at $1,688,000; the propelling installation to be of the Parsons marine type.

Both contractors were required to guarantee, under penalty, and in addition to various other trials, a successful sea trial of four hours' duration at an average speed of not less than 24 knots on a displacement of not less than 3,750 tons.

A summary of the important data of the propelling machinery of the three vessels follows: (For number and dimensions of auxiliaries, see table S.)

BIRMINGHAM.

The engines, which turn the propellers outboard, are of the vertical inverted 4-cylinder, direct-acting, triple-expansion type, with unjacketed cylinders, placed in two watertight compartments, and operating twin screws. Each engine was designed for an indicated horsepower of 8,000 at 200 revolutions per minute. with a steam-chest pressure of 250 pounds per gauge. Beginning forward, the order of the cylinders is, forward L.P., H.P., I.P., and after L.P. The forward L.P. and H.P. cranks are opposite, as are the I.P., and after L.P., the second pair being at right angles with the first. All main valves are of the piston type, worked by double-bar Stevenson links. There is one piston valve for the H.P. and two each for the I.P. and L.P. cylinders.

Engine framing is of forged-steel, cylindrical columns, trussed by forged-steel stays. All crank, thrust, line and propeller shafting is hollow, and shafts, piston and connecting rods, and working parts generally, are of forged steel:

Cylinders.

Number for each engine	4
H.P. diameter, inches	28¼
I.P. diameter, inches	45
F.L.P. diameter, inches	62
A.L.P. diameter, inches	62
Diameter of piston rods, inches	6
Stroke of all pistons, inches	36

Boilers.

The arrangement of boilers, which are of the Fore River design on the *Birmingham* and *Salem,* and of the Normand type on the *Chester,* are shown in plan on plate 3. In the Fore River type, it will be noted that the products of combustion leave at the back of the furnaces, on their way to the uptakes, and thus make a double passage across the tubes; in the *Chester's* boilers, only a single pass is made, the gases entering the tubes near the front of the furnaces, as shown on plate 4.

The closed fireroom system of forced draft is used, there being six independent Sturtevant fans installed, two to each boiler compartment, 66 inches in diameter, 24-inch width at tips of blades, of the double-inlet type, with 42-inch intakes. Each fan is directly driven by a double-acting, 2-cylinder, simple, horizontal engine (6 by 6) with cranks at 180 degrees, the steam admission and emission for both cylinders being controlled by one piston valve.

SALEM.

The *Salem* is driven by outboard-turning twin screws, the propelling machinery consisting of two Curtis marine reversible impulse turbines, one on each shaft, designed to develop 8,000 B.H.P., at 350 revolutions per minute with 250 pounds (gauge) steam-chest pressure. The machinery is ar-

ranged in two water-tight compartments, separated by an athwartship bulkhead, the starboard screw being operated by the forward turbine.

Each turbine has a pitch diameter of rotating wheels of about 120 inches (center to center of buckets), and for the ahead motion, consists of seven stages; each wheel is fitted with three rows of buckets, except in the first stage, which contains four rows. For backing, there are two stages, the wheels or rotors being mounted on the shaft in the same casing as the go-ahead stages.

Each turbine consists of a cast-iron cylindrical shell (for strength, cast-steel is used in construction of the casing of the two first stages) divided by dished cast-iron or cast-steel diaphragms into separate compartments or stages. The moving buckets are mounted on the periphery of separate wheels in each stage, and all wheels are secured to and carried by a hollow steel shaft extending the length of the turbines. Where this shaft passes through the diaphragms, bronze bushings are provided, with small clearances, in order to prevent appreciable steam leakage from stage to stage, and where the shaft extends through the heads carbon-packed gland boxes are fitted to prevent steam leaking out at the ahead end, or air leaking in at the back end. These boxes are supplied with live steam between carbon packing, which makes an effective seal against air leaking in at the back end, and which, unless checked, would cause the vacuum to be lowered. Gland-box packing, on the ahead end, is required to withstand the pressure existing in the first stage only, in order to prevent steam leakage into the engine rooms; and, as this pressure is never excessive, under any condition of operation, no serious practical difficulty arises in guarding against such leakage. A drain, or "leak off," from these gland boxes leads to the fourth stage. The ahead steam chest contains twenty expanding nozzle openings to the first-stage wheel, seventeen of which are controlled by valves, the three remaining openings being without valves and consequently always open. When in operation, sufficient nozzle valves are opened to give the de-

sired speed, the throttle valve being left wide open in order to secure full pressure in the steam chest. Variation in speed within limits may be made by throttle-valve regulation, but usually this is accomplished by opening or closing of nozzle valves.

Steam chests are of separate steel castings attached to each turbine-casing head. The astern chest contains 25 nozzles of about the same total area at the throat as the ahead chest, but only eight are provided with valves. In maneuvering, nozzle valves are left open and speed controlled by throttle valves.

First-stage nozzles are of the expanding type, for the steam in its passage to the first revolving wheel undergoes too great a drop in pressure to permit the use of the parallel-flow type, which latter is used for all other stages, the drop in pressure being much less. Diaphragm plates have openings cast in them to allow passage of steam through to the nozzles.

Drain pipes are fitted between each stage and the succeeding one, in order that condensed steam in any stage may pass to the next one of lower pressure. The exhaust chamber drains to the condenser, and the discharge is assisted by a small steam ejector. Generally, however, this ejector is used only in getting underway, being closed off when the turbines have assumed normal working conditions.

CHESTER.

The propelling machinery of the *Chester*, designed for a working steam pressure at the turbines of 250 pounds (gauge) and about 507 revolutions per minute for the contract speed of 24 knots, consists of Parsons marine steam turbines, driving four independent shafts, each shaft being fitted with one propeller. The installation comprises six ahead turbines in all, two of which—the H.P. and I.P. cruising—are for use at low powers in order to secure economy. There are also two backing turbines, located in each of the L.P. casings, and it is to be observed that this arrangement permits reversal only of the two inboard shafts in maneuvering.

Referring to plate 2, it will be seen that the outboard shafts (Nos. 1 and 4) are operated entirely by main H.P. turbines; the starboard inboard (No. 2) by the L.P. (starboard) turbine and H.P. cruising turbine, and the port inboard (No. 3) by the L.P. (port) turbine and the I.P. cruising.

The turbines may be used in any of three ways in propelling the vessel: First, for speeds up to about 18 knots, all six turbines are put in operation, since this combination results in a smaller quantity of steam being required than with either of the other two combinations. Steam is then admitted initially into the H.P. cruising turbine, exhausts into the I.P. cruising turbine, and from the latter splits and is conveyed through separate pipes to each of the main H.P. turbines. From these latter, steam is exhausted into the L.P. turbines, and finally into the condensers. Second, for speeds beyond the capacity of the 6-turbine combination, and up to about 23 knots, the 5-turbine combination is used. Steam is admitted initially to the I.P. cruising turbine, passing thence to the two main H.P. turbines, and from each of them to the connected L.P. turbine. The H.P. cruising turbine revolves idly in a vacuum. Third, for highest speed, only the four main turbines are used, steam being admitted initially to each main H.P. turbine, exhausted into the L.P.'s and then into the condensers. Both cruising turbines revolve idly in a vacuum, or may be disconnected by means of shaft couplings. It is to be understood, of course, that the 4-turbine combination may be used for any speed up to the full power, just as the 5-turbine combination may be employed for any speed up to its limit.

In any of the three arrangements, increase or reduction of power is effected by throttling, though in the 6-turbine combination, a by-pass is fitted between the first and second expansion which may be used, within limits, for this purpose. All maneuvering is done with the four main turbines only, and in event of either the 6 or 5-turbine combinations being in use when a signal is received to go astern, a quick shift is necessary to cut out cruising turbines. This simply involves

closing and opening proper valves, and need not require a greater interval of time than is usual in reversing machinery of the reciprocating type.

For the removal of water, all turbines have a connection to a system of drain piping leading to the main condensers. The outlet from each turbine is controlled by a valve which is always opened when warming up, and also on the cruising turbines when underway in case these are not in use for propulsion.

Shaft gland boxes (twelve in all) are bolted to forward and after casing heads of each turbine. In order to prevent leakage, a steam pressure of about one pound above the atmosphere is usually maintained in gland boxes when cruising, through a system of piping installed for the purpose.

As shown on plate 2, the machinery is installed in two compartments, separated by an athwartship watertight bulkhead, the two cruising turbines, in addition to the starboard main H.P. and L.P. being located in the forward engine room. For the ahead motion the inboard shafts turn outboard, and the outboard shafts turn inboard.

Boilers.

The principal dimensions and arrangement of boilers are shown on plates 3 and 4. As previously pointed out, the gases of combustion make but a single pass across the tubes in their exit to the uptakes. It is to be noted, furthermore, that, although the grate surface is the same in the boiler installations of all three vessels, the *Chester's* boilers contain but 32,040 square feet of heating surface, as against 37,992 square feet in the boiler plants of the *Birmingham* and *Salem*.

The system of forced draft used on the *Chester* is similar to that on the other scout vessels. It is of the closed-fireroom type, there being two blowers (six in all) 84 inches in diameter for each boiler compartment. Each blower is driven by a vertical two-cylinder (5 by 5) simple double-acting steam engine, with one piston valve to each cylinder. The blowers are arranged in pairs near the fore-and-aft center line of ship, with a connecting shaft between each two fans, so that both

fans run at the same speed, and in event of accident to one engine the remaining engine may be utilized to operate both fans.

Machinery Weights.

Weight of propelling machinery, shafting, bearings and propellers is as follows:

	Tons.
Birmingham (2-shaft reciprocating-engine arrangement)	234.49
Salem (2-shaft Curtis turbine arrangement)	254.80
Chester (4-shaft Parsons turbine arrangement)	207.38

Weight per horsepower (the maximum power of main engines or turbines for two consecutive hours during either steam or coal consumption trials) is:

	Pounds.
Birmingham (16,134), per I.H.P.	32.55
Salem (18,070), per S.H.P.	31.58
Chester (20,004), per S.H.P.	23.22

The weight of appendages (auxiliaries) to the above propelling machinery, including main condensers, air pumps, circulating pumps and dry-vacuum pumps or vacuum augmenters is as follows:

	Tons.
Birmingham	45.13
Salem (fitted with dry-vacuum pumps)	67.63
Chester (fitted with vacuum augmenters)	58.19

The boilers, fittings, smoke pipes and uptakes weigh as follows:

	Dry. Tons.	Wet. Tons.
Birmingham (12 Fore River boilers, 4 smoke pipes)	262.51	294.14
Salem (12 Fore River boilers, 4 smoke pipes)	262.61	294.24
Chester (12 Normand boilers, 4 smoke pipes)	272.70	307.01

Weight of boilers with fittings and water in pounds per square foot of heating surface is:

	Pounds
Birmingham	13.10
Salem	13.09
Chester	17.64

The total weight of machinery installation, including propelling machinery and appendages, auxiliary machinery, piping, boilers and fittings, smoke pipes and uptakes, lagging and clothing, flooring, ladders and gratings, fittings and gear, stores, tools and spare parts carried on board, and pipes, etc., connecting to machinery not under the cognizance of the Bureau of Steam Engineering, is as follows:

	Dry. Tons.	Water. Tons.	Total. Tons.
Birmingham	760.82	53.15	843.97
Salem	853.85	55.11	908.96
Chester	735.87	64.82	800.69

The weight per horsepower (main propelling machinery only) of all machinery (wet) is:

	Pounds.
Birmingham (16,134), per I.H.P.	117.18
Salem (18,070), per S.H.P.	112.68
Chester (20,004), per S.H.P.	89.67

Measuring Apparatus.

Water-measuring apparatus installed for the tests, together with necessary lift pumps and piping, is shown in outline on plate 5. For the propelling machinery the apparatus is composed essentially of two cylindrical tanks, one for each engine room, divided by a central diaphragm. These tanks were located on the upper deck and emptied by gravity into two distributing tanks situated immediately below on the engine-room grating. Water from each main condenser was pumped into its own feed tank in the usual way and from there elevated to the measuring tanks by means of lift pumps. It is to be noted that the central diaphragm of the measuring tanks served to divide each into two separate, similar compartments, thus, in reality, providing two measuring tanks of about 6,100 pounds capacity for each engine room. These tanks were of conical construction at the top and bottom, and each compartment terminated at the top in a cylindrical nozzle 11½ inches in diameter.

After measurement the water was dropped to the dis-

tributing tanks, which were connected by an equalizing pipe, and had a combined capacity of about 24,700 pounds. Suction pipes led from distributing tanks to feed pumps, thus providing means of returning water of condensation to the boilers. Some of the first tests (steam consumption) on the *Chester* were made without any connection between distributing tanks, and this was found unsatisfactory, the tanks not being large enough to take, at all times, the full capacity of a liberated measuring tank. In addition to this, the lift pumps in the beginning gave considerable trouble, necessitating the use of one main lift pump only, and thus requiring the two main feed tanks to be connected by opening the equalizing valve. Subsequently the two distributing tanks were connected to overcome this difficulty.

Steam used by the various auxiliaries, heaters, steam traps, drains, etc., was condensed by the auxiliary condenser and the water of condensation delivered into an auxiliary tank installed as part of the test outfit. This tank, of rectangular section, was divided into two compartments, with an approximate capacity of 1,130 pounds of water each, and by means of an auxiliary lift pump, could be readily emptied as desired. The water of condensation from the auxiliaries was, after measurement, delivered into the forward distributing tank, and any make-up water required during the tests was pumped from the double bottoms directly into this tank. It should be noted that the auxiliary measuring appliances, embracing tanks, piping and lift pumps, constitute a system entirely distinct and separate from the main measuring apparatus.

The discharge from each lift pump terminated in a swinging goose neck which permitted readily shifting from one compartment to the other of each tank. The shell of the main measuring tanks was carried above the head, thus forming an open-topped receptacle in which any overflow from the measuring tanks was caught and returned to the feed tanks by means of drain pipes.

Plate 6 shows sectional views of main, distributing and auxiliary tanks.

The dynamo plant, composed of three 32-kilowatt units, was provided with its own condenser and pumps, and the air-pump delivery was so arranged that the water of condensation could be delivered either directly into the auxiliary measuring tank or to the E.R. auxiliary condenser. As the auxiliary exhaust line of piping was also connected to the dynamo plant, it was possible to utilize the dynamo condenser to assist in condensing the steam from the auxiliaries in operation in case the auxiliary condenser became too small in condensing surface to properly handle the auxiliary exhaust steam at high powers.

Calibration of Measuring Tanks.

All measuring tanks, prior to installation on shipboard, were carefully calibrated on platform scales at the Boston Navy Yard. Each tank was filled with water at 100 degrees temperature and the weight of contained water ascertained for each inch rise in height. Glass water gauges, graduated brass scales, and thermometers were attached to each tank for use during steam consumption trials.

I. Standardization Runs.

Dates of dry docking, etc.

Speed performances of vessels are affected by state of wind and sea, and especially by condition of underwater hull as to cleanliness. Reduction in speed invariably follows sea-growth accumulations on ships' hulls, and in order to minimize the uncertain and varying influence attributable to this, during comparative sea trials of the scouts, it was the aim to run all such tests with ships' bottoms clean. This plan, however essential to accurate comparison, proved a most difficult one to execute with entire satisfaction, and, no doubt seemingly small, inexplainable discrepancies of speeds and power on some of the various trials are in a large measure due to this cause.

As a preliminary to standardization runs, the three vessels were dry docked and the bottoms cleaned and painted with

the same variety of (McInnes) paint. Each vessel was standardized over the Rockland measured-mile course just prior to undertaking (II) steam-consumption tests, and as will appear later, this involved standardizing the *Salem* on two different occasions, as two series of steam-consumption tests were carried out. In these trials, moreover, and as well as all other (III and IV) sea runs, the aim was to load each vessel so that the displacement for each trial should be as nearly as could be calculated in the beginning, an average of 4,000 tons.

The following table (A) is compiled from results of standardization runs:

TABLE A.—*Revolutions and shaft horsepower from standardization curves.*

Speed, in knots.	*Birmingham.*		*Salem.*				*Chester.*					
			1st series.		2d series.		4-turbine.		5-turbine.		6-turbine.	
	R.p.m.	*Shaft H.P.	R.p.m.	Shaft H.P.	R.p.m.	Shaft H.P.	R.p.m.	Shaft H.P.	R.p.m.	Shaft H.P.	R.p.m.	Shaft H.P.
10.0	74.2	902	...	...	139.2	870	203.0	790	203.0	630	203.0	700
10.5	77.7	1,010	147.3	720	146.3	1,000	214.0	900	214.0	750	214.0	800
11.0	81.4	1,137	153.8	830	153.8	1,140	225.0	1,050	225.0	890	225.0	940
11.5	85.1	1,278	161.0	980	161.0	1,300	236.0	1,200	236.0	1,030	236.0	1,100
12.0	88.8	1,410	168.0	1,140	168.0	1,460	246.7	1,380	246.7	1,220	246.7	1,280
12.5	92.5	1,579	175.0	1,330	175.0	1,650	258.0	1,580	258.0	1,430	258 0	1,500
13.0	96.5	1,795	182.2	1,520	182.2	1,860	268.5	1,800	268.5	1,670	268.5	1,730
13.5	100.2	2,012	189.3	1,740	189.4	2,080	279.5	2,040	279.5	1,900	279.5	1,980
14.0	104.0	2,237	196 4	1,980	196.8	2,350	290.5	2,300	290.5	2,180	290.5	2,290
14.5	108.0	2,491	203.5	2,225	204.0	2,620	301.5	2,570	301.5	2,470	301.5	2,620
15.0	111.8	2,763	210.8	2,500	211.1	2,910	312.5	2,870	312.5	2,800	312.5	2,980
15.5	115.7	3,055	217.8	2,780	218.3	3,225	323.5	3,200	323.5	3,110	323.5	3,350
16.0	119.7	3,365	224.8	3,100	225.8	3,580	334.0	3,550	334 0	3,500	334.0	3,750
16.5	123.5	3.703	231.9	3,430	233.0	3,960	345.5	3,930	345.5	3,900	345.5	4,180
17.0	127.5	4,070	239.0	3,800	240.1	4,370	356.0	4,320	356.0	4,320	356.0	4,630
17.5	131.4	4.493	246.0	4,160	247.3	4,800	367.0	4,780	367.0	4,820	367 0	5,100
18.0	135.3	4,860	253.2	4,580	255.0	5,280	378.0	5,220	378.0	5,300	378.0	5,600
18.5	139.3	5,301	260.2	5,000	262.2	5,780	389 0	5,700	389.0	5,850	389.0	6,150
19.0	143.2	5,771	267.5	5,450	269.8	6,330	400.0	6,250	400.0	6,400	400.0	6,700
19.5	147 2	6,270	274.8	5,930	277.0	6,880	410.0	6,780	411.0	7,000	...	...
20.0	151.3	6,834	282.0	6,470	284.4	7,480	421.0	7,400	422.0	7,650	...	...
20.5	155.8	7,398	289.0	7,000	292.0	8,130	432.0	8,030	433.0	8,280	...	...
21.0	159.5	8,018	296.5	7,600	299.6	8,800	442.5	8,700	444.0	9,050	...	...
21.5	163.7	8,694	303.9	8,220	307.0	9,500	454.0	9,500	455.0	9,800	...	...
22.0	168.0	9,428	311.2	8,900	314.8	10,270	465.0	10,430	466.0	10,570	...	...
22.5	172.5	10,226	318.9	9,640	322.4	11,100	476.0	11,200	478.0	11,480	...	...
23.0	177.5	11,213	326.8	10,475	330.4	12,000	488.0	12,180	490.0	12,420	...	...
23.5	183.1	12,408	335.0	11,430	338.8	13,040	501.5	13,330	504.0	13,600	...	...
24.0	189.5	13,912	344.0	12,600	347.3	14,250	516.0	14,650	...	...	...	...
24.5	197.2	15,837	353.8	14,050	357.0	15,730	533.5	16,380	...	...	...	...
25.0	...	...	364.6	16,000	367.0	17,550	554.0	18,450	...	...	...	...
25.5	...	...	...	...	...	...	580.0	21,280	...	...	...	...

* Assumed to be 94 per cent. of indicated horsepower.

Standardization of the *Chester* involved running over the measured-mile course using all three—4, 5 and 6—combinations of main propelling machinery. It was found, however, that up to about 19 knots, although there was slight variations in horsepower, one speed-revolution curve answered practically for all turbine combinations; above that speed, and for the 4 and 5 combinations, separate speed-revolution curves were laid down.

Prior to (III) coal-consumption trials, all vessels were again docked and bottoms painted. The number of days intervening between date of docking and commencement of the various tests is shown below:

.....	Days out of dock.			
	Birmingham.	*Salem.*		*Chester.*
		First.	*Second.*	
Before standardization trials.....	80	12	33	49
commencement of main steam consumption tests.	104	27	29	61
first coal-consumption test.....	4	6	...	8

It will be observed from table A that although the two standardizations of the *Salem* (first and second series) were run (December 15, 1908, and August 11, 1909) at the same displacement, results as to revolutions and power for a given speed differ somewhat. This is probably attributable to unlike weather conditions and different condition as to cleanliness of underwater hull when the trials were run. There is likelihood, also, that torsion-meter readings were unreliable, or a further possibility of inaccurate determination of zero points of the instruments, although investigation seems to show slight probability of large differences of power being accounted for in this way.

Plates 8, 9 and 10 show standardization data for the three vessels, and on the same sheets for comparison, points have been plotted for power, speed and revolutions, as determined on later (steam and coal-consumption) sea trials.

II. Steam-Consumption Trials of Main Propelling and Auxiliary Machinery.

Of the various trials undertaken, those of this group, comprising tests to fix steam consumption of the propelling machinery of the three vessels at different sea speeds, are believed to be of most importance. The object was to establish comparative steam expenditures of the three types of main propelling installations, progressively, from about 10 knots to maximum speed, and, incidentally, this involved determination of the steam used collectively on each trial by the auxiliaries in operation. It will be apparent that in running these trials, it was necessary to keep in view, and, as far as possible, guard against any features which might have an indeterminate influence on speed, such as the following: (1) Unlike conditions of underwater hull as to cleanliness; (2) variable weather and sea conditions; (3) dissimilarity in displacement.

All vessels were fitted with separate and similar water-measuring apparatus, and the series of tests on each were undertaken at different times, as the ships could be spared for the purpose. When once started, however, the entire series was finished before assignment of the vessel to duty elsewhere, and for convenience in coaling as the trials progressed, the coaling plant at Bradford, R. I., was selected as an accessible base from which to operate. For the second series of tests on the *Salem,* the coaling station at Frenchmans Bay was chosen as a base.

It was the aim to have the bottoms of all vessels free from marine growth, and as far as could be judged from the condition of that part of the underwater body visible at the water line there was no radical difference in this respect when the trials were run. The time interval between coming out of drydock and commencement of these trials, however, was not, as has been previously pointed out, the same for any of the three vessels.

Tests were run during favorable weather only; that is to say, under conditions of smooth sea and comparatively light

winds, and, as far as practicable, in day time, the ship always coming to anchor about nightfall. Frequent delays occurred by reason of unfavorable weather, and often, on slow-speed runs, advantage was taken of the protected waters of Long Island Sound in order to escape unsuitable weather conditions prevailing in the open sea.

An effort was made to maintain uniformly a standard mean displacement of 4,000 tons on all trials, but evidently this could be carried out practically within reasonably narrow limits only. The draught was adjusted by water ballast (using trimming tanks and double-bottom compartments) at the beginning of each day, when tests were to be made, so that the resulting displacement would be slightly over the figure stated, due consideration being given to the amount of coal likely to be consumed. The displacement for each trial run during the day was readily figured from the draught on coming to anchor for the night. As the ship's displacement gradually decreased, due to daily fuel consumption, a condition was finally reached, after completion of a number of tests, where it was impracticable to arrange for the standard displacement by water ballast alone; at such times tests were temporarily discontinued until the ship could be again coaled.

A brief description has been given of the water-measuring appliances, and it will only be necessary to set forth here more in detail the method followed in its operation. Reference has been made to lift pumps which were installed as part of the test outfit. Three such pumps were provided on each vessel, one in connection with each main feed tank, and one for the auxiliary measuring tank. The sole function of the main lift pumps was to draw water from the feed tanks and elevate it to the measuring tanks located on the spar deck. From these latter, after measurement, the water was dropped by gravity into distributing tanks below, from which, through proper connections to the feed pumps, the boilers were supplied with feed water. The auxiliary-lift pump handled all water, after measurement from the auxiliary tank, and delivered it to the forward distributing tank; water of con-

densation from auxiliaries in operation was discharged into auxiliary measuring tank by the air pump of the auxiliary condenser.

It was the usual practice, after the propelling machinery had been regulated to the number of revolutions desired (as determined from the standardization speed-revolution curve) and with the water of condensation passing through both main and auxiliary measuring apparatus, to allow about one-half hour to intervene before beginning to record data, in order to guard against irregularities of operation and for the further purpose of permitting the machinery to assume normal working conditions. Eight observers were stationed as follows: one at each main measuring tank; one at the auxiliary measuring tank; two in each engine room to record data of counters, gauges, revolutions, etc., of auxiliary machinery, and height of water in feed tanks; one to record data of auxiliaries outside of the engine-room spaces. In addition to the above, a requisite number of observers were assigned to secure data from which the power of the propelling machinery could be calculated.

A stand-by signal was given one minute before each test began, generally by engine-room gongs; also at the instant of beginning each test, and thereafter at precisely twenty-minute intervals as the test progressed. At the moment of starting, measuring-tank observers shifted goose-neck connections, and thus directed the water of condensation into the empty compartment of the measuring tanks, the valves at the bottoms controlling the outlets having been previously closed. As each measuring-tank compartment filled, the remaining one was emptied and prepared for the reception of water, so that both compartments were used alternately as the test proceeded. A record was kept of the elapsed time in filling each compartment, as well as of the temperature of the water.

In the engine room the height of water in each feed tank at all time intervals during the tests was recorded, a graduated scale being attached to facilitate making such entries. These tanks, moreover, had been calibrated, and although the aim

in all tests was to maintain the water delivered into them at a constant height, in event of departure from this, the exact weight of exhaust steam condensed by each main condenser could be figured at any time by applying a proper correction. The feed tanks were also connected by an equalizing pipe, making it possible to use but one main measuring tank and its lift pump for both engine rooms; and it may be noted in passing that during the first tests on the *Chester* the forward lift pump gave considerable trouble from unsatisfactory working, unavoidably necessitating on such tests connection of the two feed tanks through the equalizing pipe. Ordinarily, however, the tanks were disconnected, as this method furnished the means of separately determining the weight of exhaust steam entering each main condenser.

Steam-consumption tests may be classed under two general headings, as follows: First, those in which none of the auxiliary exhaust steam was used in the main propelling machinery to assist propulsion; and, second, those in which all available auxiliary exhaust, not utilized in the feed heaters, was conveyed to the propelling machinery to assist propulsion. Otherwise stated, the two classes comprise tests, (1) in which exhaust steam from the auxiliaries was led into the feed-water heaters, afterwards being delivered to and condensed in the auxiliary condenser, and (2) tests in which the auxiliary exhaust line was open, not alone to the feed heaters, but as well to L.P. engine receivers (*Birmingham*) or some stage (*Salem* and *Chester*) of main turbines. In tests of the latter character only condensed exhaust from dynamos, trap drains, etc., delivered into the auxiliary measuring tank, the regulating valve in auxiliary exhaust line at the auxiliary condenser being tightly closed to prevent entry therein of exhaust steam. It will be evident that water of condensation collected and measured in the main tanks in such tests included steam used by the main propelling machinery, and in addition part of that used by the auxiliaries. A pressure above the atmosphere was always maintained in the auxiliary exhaust line, which was affected by a regulating valve on the auxiliary

condenser in tests of the class first mentioned, and by manipulation of proper valves attached to propelling machinery on other tests. It should be noted that the larger the percentage of auxiliary exhaust steam delivered to the main engines or turbines the greater the speed and power developed, the highest speed and power on a given steam consumption being obtained when the feed heaters are not in use. The apparent economy, as measured in total steam per knot and per horsepower, would then reach a maximum. This, however, does not represent the point of maximum economy when speed and power are based upon coal consumption, for the use of sufficient auxiliary exhaust in the feed heaters to give the feed water the highest temperature increases the amount of steam evaporated in the boilers when the same amount of coal is being consumed, so that the increased output more than counterbalances the apparent loss due to the use of exhaust steam in feed-water heaters. This fact should always be taken into consideration when the performance of any vessel is given in terms of total water per knot or per horsepower, otherwise misleading inferences may be drawn. It may be further pointed out in connection with steam-consumption trials that close regulation of auxiliaries to the requirements (which to a large extent are dependent upon speed of vessel) was not attempted. In this respect data of coal-consumption trials furnish a safer guide as to the least quantity of steam necessary for operation of the auxiliaries, since on all ships during these trials every effort was made to reduce this expenditure to a minimum.

Power of the main propelling machinery was calculated, for the *Birmingham,* from indicator cards. Torsion meters, of the Denny-Johnson type, were fitted on the *Salem* and *Chester* to obtain shaft horsepower, determinations with these instruments being made by comparison of the torsion on a length of turbine shafting, which before installation had been carefully calibrated, and torsion readings on the same length of shaft during the trials. In the latter, the torsion meters measured the torsion by means of magnets, electrical resistance

coils, etc., readings depending for accuracy upon adjustments for sound disappearance. Particular care was exercised, at all times, to secure accurate torsion-meter readings, but the type mentioned as installed on these vessels is believed to be too delicate, both in construction and theory of operation, for satisfactory and reliable results on shipboard. Moreover, at low powers, because graduations for sound disappearance were so widely apart, a source of considerable possible error existed, no matter how carefully readings were taken, as illustrated by the following: Scale graduation, between adjacent contact points on the instrument, progress by 0.02 beginning at 0. The smallest reading recorded on any test was 0.07, which lies between the contact points of 0.06 and 0.08, and which, therefore, can only be regarded as an approximation. Furthermore, this reading was subject to correction for determination of zero point of the instrument, which is established by steaming ahead, for example, on the port screws, while the starboard shafts revolve idly and presumably without torque. It will be evident that this latter reading can not be taken with any greater degree of accuracy than the former. At higher powers, the possibility of error is smaller in percentage of the total, and in addition, torsion-meter readings can be more readily noted, because at higher rotative speeds the sound is more distinguishable.

Since on various sea trials it was possible only to obtain shaft horsepower of the main propelling equipments of the *Salem* and *Chester,* while on the *Birmingham* torsion meters were not installed, the necessity of selecting either shaft or indicated horsepower as the unit for purposes of power comparison is apparent. Shaft horsepower has been chosen principally because power expended in propulsion on two of the three vessels was thus measured. Sufficient experimental data is not extant to fix accurately the existing ratio between shaft and indicated horsepower. Moreover, this ratio varies somewhat for the same engine, being greater at lower than at higher powers, and besides is not constant for engines of the same power if of different types. In practice, however, it

is generally considered as constant for all usual ranges of power of the same engine. In calculations relating to these trials, a ratio of 0.94 has been assumed in converting indicated into shaft horsepower, but this should be regarded only as a close approximation. It may be further pointed out, in this connection, that no attempt has been made to convert indicated into shaft horsepower for the auxiliaries, since computations of this character introduce so many uncertainties, at best, as to render such calculations of doubtful value.

Calorimeters of the Barrus throttling type were installed and readings therefrom recorded on all sea trials. These were located between the throttle and steam chest, as follows: One in each engine room of the *Birmingham* and *Salem;* on the *Chester,* four in all, one each to H.P. cruising, I.P. cruising and main H.P. turbines.

In carrying out steam-consumption trials on main machinery the importance of obtaining uniformity in water rates, under like conditions of operation, was realized. The weight of condensed steam collected was checked at frequent intervals as the tests progressed, and each test continued until agreement of results, within reasonable limits, was shown. Almost all tests cover a time interval of at least two hours, and this is believed to be sufficiently long to secure reliable data when fluctuations in water rates are slight. Where, however, considerable variation occurred in this particular, due to unknown causes, tests were continued over a longer period.

Results of evaporative boiler tests hereinafter described show practically no moisture in the steam, even for higher combustion rates. In calculations relating to steam consumption of main and auxiliary machinery, therefore, it has been assumed that dry steam was supplied at all times. In this connection it may be remarked that steam separators were fitted in each engine room, in the main steam line of the *Birmingham* and *Salem,* but no similar appliance was installed on the *Chester.* Through an oversight in design and construction, the interior diaphragm being improperly located, the forward separators on both ships were only partly efficient

for the purpose intended, unless cross-connection pipe between port and starboard main steam lines in forward engine room was not used. As installed, and with valves open on the pipe mentioned, steam from the port boilers could pass directly to the forward propelling engine, the separator merely serving as a means of communication. The first series of steam-consumption trials on the *Salem* were run with this cross-connection pipe in use, and on some tests violent fluctuations were noticeable of the lower calorimeter thermometer. This may be attributed to the fact that the forward turbine on all these tests, when making the same number of revolutions, used more steam than the after, and consequently the supply was no doubt drawn partially through this cross-connection pipe. In later trials, connection between port and starboard main steam lines was effected through piping in the firerooms only.

The following was the order in which steam-consumption tests of the main propelling machinery took place: *Chester,* November 18 to December 5, 1908; *Salem* (first series), December 31, 1908, to January 3, 1909; *Birmingham,* January 7 to 12, 1909; *Salem* (second series), August 7 to 14, 1909.

BIRMINGHAM.

[Table G.]

The auxiliary exhaust line of piping on this vessel is connected to the second receivers of main engines. Tests Nos. 2, 4, 7 and 9 were carried out with receiver valves slightly open, the object being to thus utilize any auxiliary exhaust steam not condensed in the feed-water heaters. It will be apparent that, by cutting out the latter, all exhaust from the auxiliaries becomes available to assist in propulsion, but necessarily this involves lowering the boiler feed-water temperature; and while in this way a less number of pounds of steam per unit of power developed unquestionably results, no increase in economy, based upon coal consumed, is shown. Receiver valves were so regulated in these tests, therefore, as to maintain feed-water temperature at about 150 degrees.

Tests Nos. 1, 3, 5, 6, 8 and 10 were conducted with engine receivers valves closed and engine-room auxiliary condenser in operation to care for such excess of auxiliary exhaust steam as was not condensed in the feed heaters.

SALEM.

[Tables H and I.]

As previously pointed out, two series of steam-consumption tests were carried out on the *Salem*. The second series, however, was not made until after completion of (III) coal-consumption tests and (IV) full-power run.

On all sea trials of this vessel it had been noticed that the starboard turbine showed less efficiency than the port, inasmuch as more nozzles were required to be open on the former (with the same steam-chest pressure) to maintain the same revolutions of the two turbines. To ascertain the cause of this difference, it was decided, after the full-speed run of twenty-four hours, to lift top casings of both turbines. Upon examination it was found that the starboard turbine had sustained damage by lodging of an obstruction between the 5th stage nozzles and buckets, resulting in almost completely closing up steam passages through the first row of buckets of that stage, and which accounted for decrease in revolutions of the starboard as compared with the port turbine. It was apparent, also, that both turbines were generally in a bad condition, due to excessive working of rotor shaft in a fore-and-aft direction, which was made possible by yielding of thrust foundations. In both turbines the rotor and shaft had worked fore and aft until the clearances in most of the stages between rotor buckets and intermediates had not only disappeared, but the rotor-bucket bases and shrouds had rubbed and cut into the bases and shrouds of intermediates until the latter had, in many instances, been reduced in width from $\frac{1}{8}$ to 3/16 of an inch. Moreover, while in the port or after turbine the movement of the rotor had been forward, yielding to the thrust of the screw and the steam thrust which act in the same direction in that turbine, in the forward or starboard turbine, the

displacement of the rotor had been aft against the screw thrust, yielding to the abnormal steam thrust brought about by the closed buckets of the 5th stage.

At the overhauling which followed this examination, and which took place at the works of the Fore River Shipbuilding Company, buckets and intermediate blading of both turbines were put in good condition, and additional braces fitted to thrust bearings; all movable valves (of the slide type) between stages, originally installed over diaphragm-nozzle openings, were also removed. These latter were designed and fitted with a view of controlling to a limited extent the pressure and consequently the flow of steam in passing through the turbines. They were spaced at intervals circumferentially, and arranged to be hand operated through a stem extending to the outside casing. It was believed that by regulation of the steam flow through the various stages at low powers, by means of these valves, steam economy might thus be effected. Practically, however, as demonstrated during the first series of tests (Nos. 3 and 5) there appeared to be no material difference in steam economy, no matter whether these valves were closed or open.

Upon completion of repairs, the *Salem* was ordered on a cruise to the Canary Islands, later returning to the Boston Navy Yard for drydocking and reinstallation of water-measuring apparatus before commencement of the second series of steam-consumption trials. The first (table H) series consists of eight and the second (table I) of eleven tests.

Particular attention is invited to results of tests Nos. 2 and 5 of the second series, which were run at about the same speed of the vessel. Test No. 5 was made with all (20) nozzle valves open, regulation being by turbine throttles entirely, for the sole purpose of comparing steam consumption under these conditions with that in test No. 2, in which but 3 (after) and $3\frac{3}{16}$ (forward) nozzles were open. In the latter test (which is the ordinary method of operation) turbine throttles were practically wide open, slight modifications in revolutions when necessary being made by manipulation of the throttle valves. It is to be noted further that there was wide difference

in steam-chest pressures in the two tests, the average (per gauge—both turbines) for No. 2 test being about 210 pounds and for No. 5 test about 30 pounds, while pressures in main steam line were approximately 249 and 201, respectively. As will be seen from an examination of the data, there was no appreciable difference in economy of the turbines in the two tests. Suitable connections were installed to the third and fourth stages of main turbines for utilization of auxiliary exhaust steam. In test No. 4 (first series) and tests Nos. 3, 4, 7 and 9 (second series) the auxiliary exhaust line was open to feed heaters as well as to third stages of turbines.

CHESTER.

[Tables J, K and L.]

Steam-consumption tests on the *Chester,* numbering twenty-nine in all, embrace trials of the 4, 5 and 6-turbine combinations at various speeds. It should be pointed out that the results of the first five are at variance with later similar tests in the series, and are regarded as unreliable. These were the first of the steam-consumption tests to be undertaken, and it is probable that inaccuracies resulted from inexperience in operating the main machinery in connection with measuring apparatus, the difficulties of which, in the beginning, were greatly accentuated by unsatisfactory working of main lift pumps.

Economy tests on turbines of the Parsons type are influenced to a great extent by the amount of clearance allowed between dummy pistons and rings which controls steam leakage. On the *Chester* adjustments of dummy clearances are made by movement of the lower and upper halves of thrust blocks, and the operation is one requiring both skill and accuracy. Upper half of thrust blocks are intended to take the steam thrust only, while the lower half absorbs the propeller thrust. For the two cruising turbines, which are on inboard shafts (Nos. 2 and 3) and located forward of each main L.P. turbine, expansion couplings are installed, the arrangement being such that thrust bearings of these turbines are not

subjected to propeller thrust. All main turbines are designed to be balanced, the end thrust on the rotor and the unbalanced thrust on the blading, both resulting from steam pressure, being equal to and in the opposite direction to the propeller thrust. Perfect equilibrium, however, under all conditions of operation is practically impossible, and while for some speeds and turbine combinations propeller thrust predominates, others occur in which the steam pressure overbalances the propeller thrust. It will be apparent that in either case there is a tendency to a fore-and-aft movement (technically known as "float") of shaft and its attached rotor, which is forward in event of propeller thrust preponderating and aft under reverse conditions, or when steam thrust overbalances propeller thrust. The amount of "float," although small in any case, bears directly upon dummy clearances, operating to enlarge or reduce the same, depending upon whether the shaft movement is forward or aft. No effort was made while steam-consumption trials were in progress to adjust dummy clearances closer than usual service practice, since it was the aim to obtain steam economy under ordinary cruising conditions and not ideal results.

Brief mention has been made in describing the *Chester's* turbines of the means employed to prevent leakage (in or out) around shafts where the latter pass through turbine-casing heads. Plate 7 shows sectional views of the H.P. and L.P. turbine gland boxes, from which, it will be observed, that the packing consists, essentially, of a number of rows of knife-edge strips, supplemented by brass packing rings. The function of gland boxes is to check leakage of steam into the engine rooms when the pressure is greater in the rotor casings than the atmosphere, or air leakage into the turbines in event of the pressure being below the atmosphere, which latter, unless prevented, would cause a noticeable vacuum drop. Conditions arise, with the various turbine combinations, such that certain gland boxes (8 of the 12 provided) are required to pack against a steam pressure out, while under other conditions of operation these same gland boxes

must be practically tight against an air leakage in. In operating the 6-turbine combination, to cite an example, gland boxes of cruising turbines are required to pack against a steam pressure in rotor casings, which usually is above the atmosphere. When, however, the 4-turbine combination is in use, both cruising turbines revolve idly in a vacuum (drains to these turbines being open to the main condenser) and their gland boxes are required to be tight against a leakage of air into the turbine casings. Generally speaking, gland boxes of initial turbines are required to resist steam leakage out, but in all cases the function of the gland boxes of L.P. turbines is invariably to prevent air leakage into the casings.

Glands of all turbines are connected by branches to a system of piping, with suitable cut-out valves extending the length of both engine rooms. When steaming, it is the aim to maintain in this system at all times a pressure of about one pound per gauge, and this may be effected through a branch connection to the auxiliary exhaust line, or in case steam leakage out from any gland box is considerable in amount use of such steam is made for this purpose. With the type of gland box installed on the *Chester* steam leakage depends largely upon the highest final pressure existing in any turbine casing, and at full power, with the 6-turbine combination in use, this pressure becomes a maximum, being approximately the same as the initial pressure in the I.P. cruising turbine. Under such circumstances (and as well other powers of the 5 and 6-turbine combinations) the gland leakage is more than sufficient to pack the entire gland system, and unless provision were made to dispose of such excess steam an intolerable condition would soon result in the forward engine room. A 3-inch branch pipe, with cut-out valve, connects the gland system with the main condenser, and through this any surplus leakage is delivered. It will be noted that steam thus condensed is not fully utilized in the turbines, and from the standpoint of economy it would seem that a branch connection from the gland system to one or more stages of the low-pressure turbines would be a desirable addition.

TABLES OF DATA AND PERFORMANCE CURVES.

On account of the large number of steam-consumption tests carried out, the data are too voluminous to print in full. As illustrating the method followed, however, one test on each ship—at approximately 15-knots speed—is shown in detail in tables D, E and F. A summary of all tests is given in tables G, H, I, J, K and L.

Performance curves, plates 11 and 12, have been drawn from results of steam-consumption tests, and figures in tables B and C have been compiled therefrom. It will be seen (plate 11, table B) that, based on total steam used for all purposes (excluding lift pumps), the *Birmingham* is the most economical of the scout vessels up to (a) 20.6 knots (50 per cent. of designed power of main engines). Above that speed the *Chester*, using the 5-turbine combination, becomes the most economical and at (b) 21.6 knots the 4-turbine combination is more economical than the *Birmingham's* reciprocating engine installation. Up to (c) 22.25 knots, the *Birmingham* is more economical than the *Salem* (second series of tests), but above this speed the *Birmingham* becomes the least economical of the three vessels. Due probably to excessive gland leakage, the *Chester's* 6-turbine combination is less economical than the 5-turbine combination above (d) 17.4 knots; the 5-turbine combination, up to the limit of its speed, is invariably more economical than the 4-turbine combination. The *Chester's* 4-turbine combination is less economical than the *Salem's* installation up to (e) 19.45 knots, but more economical above that speed.

Based (pl. 12 and table C) on steam used at various speeds per hour by the main propelling machinery only (exclusive of all auxiliaries) the figures above stated change but slightly. Using reference letters as in the previous paragraph, the figures become (a) 20.6, (b) 21.5, (c) 22.45, (d) 18.0, (e) 18.9.

Curves plotted on plates 8, 9 and 10 show, in addition to other data, points for power, speed and revolutions as found on steam-consumption trials.

TABLE B.—*Total steam consumption for all purposes (exclusive of lift pumps) at speeds from 10 to 25 knots, at knot intervals.*

COMPILED FROM PLATE No. 11.

Speed.	Steam used per hour (pounds).					Steam consumption in percentage, referred to Chester's 4-turbine combination as 100.				
	Birmingham.	Salem (second series).	Chester.			Birmingham.	Salem (second series).	Chester.		
			Turbines in use.					Turbines in use.		
			Six.	Five.	Four.			Six.	Five.	Four.
10 knots	34,000	44,000	43,200	45,500	46,600	73	94	93	98	100
11 knots	39,000	51,000	48,600	51,700	53,600	73	95	91	96	100
12 knots	44,700	58,200	54,700	58,100	61,200	73	95	89	95	100
13 knots	51,000	66,200	61,000	65,000	69,200	74	96	88	94	100
14 knots	58,000	74,700	68,200	72,300	77,700	75	96	88	93	100
15 knots	66,000	84,000	76,500	80,500	87,000	76	97	88	93	100
16 knots	75,300	94,200	86,500	89,500	97,000	78	97	90	92	100
17 knots	86,200	105,400	98,500	100,300	108,200	80	97	91	93	100
18 knots	98,000	118,200	.....	111,400	120,000	82	98.5	.....	93	100
19 knots	114,400	132,500	.....	124,500	133,500	86	99	.....	93	100
20 knots	134,000	150,000	.....	138,500	148,500	90	101	.....	93	100
21 knots	160,000	170,500	.....	155,000	166,500	96	102	.....	93	100
22 knots	192,000	195,000	.....	173,500	187,000	103	104	.....	93	100
23 knots	232,800	227,400	.....	.....	214,700	108	106	.....	.....	100
24 knots	285,500	273,000	.....	.....	254,800	112	107	.....	.....	100
24.5 knots	.....	300,000	.....	.....	284,000	.....	106	.....	.....	100
25 knots	.....	329,000	.....	.....	.....	.....	.....	.....	.....	.....

TABLE C.—*Steam consumption of main engines or turbines only at speeds from 10 to 25 knots, at knot intervals.*

COMPILED FROM PLATE No. 12.

Speed.	Steam used per hour (pounds).					Steam consumption in percentage, referred to Chester's 4-turbine combination as 100.				
	Birmingham.	Salem (second series).	Chester.			Birmingham.	Salem (second series).	Chester.		
			Turbines in use.					Turbines in use.		
			Six.	Five.	Four.			Six.	Five.	Four.
10 knots	21,600	29,900	24,250	31,400	32,300	67	93	75	97	100
11 knots	26,200	36,100	30,200	36,900	38,900	67	93	78	95	100
12 knots	31,300	42,800	36,200	42,500	45,700	68	94	79	93	100
13 knots	36,900	50,000	42,500	48,800	53,000	70	94	80	92	100
14 knots	43,000	57,800	49,000	55,400	61,000	71	95	80	91	100
15 knots	50,400	66,700	56,200	62,800	69,800	72	96	81	90	100
16 knots	58,900	76,500	63,900	71,400	78,900	75	97	81	90	100
17 knots	68,800	87,300	73,900	81,100	88,900	77	98	83	91	100
18 knots	80,000	99,000	92,400	92,000	100,000	80	99	92	92	100
19 knots	94,500	112,000	.....	104,200	112,000	84	100	.....	93	100
20 knots	113,000	127,300	.....	118,000	125,900	90	101	.....	94	100
21 knots	136,500	145,200	.....	133,000	141,800	96	102	.....	94	100
22 knots	166,000	169,000	.....	150,000	160,700	103	105	.....	93	100
23 knots	203,200	199,000	.....	.....	184,800	110	108	.....	.....	100
24 knots	247,000	238,000	.....	.....	218,000	113	109	.....	.....	100
24.5 knots	.....	261,800	.....	.....	244,000	.....	107	.....	.....	100
25 knots	.....	286,400	.....	.....	.....	.....	.....	.....	.....	.....

TABLE D.—*Steam consumption tests.—Summary sheet.*

Test No. 5. Date, January 8, 1909. Duration of test, 2 hours. Speed, 14.95 knots.

U. S. S. Birmingham. STEAM CONSUMPTION PER HOUR, IN POUNDS.

.....	Steam.	Steam.	Steam.	Steam.	I.H.P.	I.H.P.	S.H.P.	Steam per hour.	
								Per I.H.P.	Per S.H.P.
(1) Main engines, (a) forward	24,329	...	...	...	...	1,451.9	1,364.8	16.8	17.8
(b) after	25,426	...	...	...	...	1,496.2	1,406.4	17.0	18.1
(2) Total for both engines (by measurement).	...	49,755	...	...	...	2,948.1	2,771.2	16.9	18.0
(3) Auxiliaries essential to the propulsion of the ship (estimated from previous tests), as follows:									
(c) Main air pumps	3,426	...	...	...	18.4	...	...	186.0	...
(d) Main circulating pumps	1,580	...	...	...	33.9	...	...	46.6	...
(e) Main feed pumps	2,300	...	...	...	31.2	...	...	73.7	...
(f) Forced-draft blowers	1,668	...	...	...	15.6	...	...	106.9	...
Total for above auxiliaries [(c) + (d) + (e) + (f)].	...	8,974	...	...	...	99.1	*99.1	90.5	...
(4) All machinery necessary for propulsion	...	...	58,729	...	...	3,047.2	2,870.3	19.3	20.5
(5) All other purposes, except lift pumps [(6) − (4)].	...	...	6,160	...	...	...	...	...	...
(6) Total for all purposes, except lift pumps	...	...	...	64,889	...	...	...	...	...
(g) Estimated for lift pumps	...	...	...	523	...	...	...	...	...
(7) Grand total for all purposes (measured)	...	...	...	65,412	...	...	...	...	...

* The I.H.P. of auxiliaries is not converted into S.H.P.

It is assumed in above table that I.H.P. × .94 = S.H.P.

NOTE.—All water and exhaust steam discharged into main condensers is delivered into main measuring tanks. Auxiliary measuring tanks take the remainder. The sum of the two is the total steam used for all purposes.

TABLE D.—*Steam-consumption tests.*—Continued.

Test No. 5. Date, January 8, 1909. Cut-off in cylinders in decimal of stroke: H.P., .60; I.P., .65; F.L.P., .50; A.L.P., .50.

U. S. S. Birmingham. FORWARD ENGINE-ROOM OBSERVATIONS.

Time.	Steam pressures. (G) = Gauge. (AB) = Absolute. In engine room (G).	In H.P. steam chest (G).	Receivers. I.P. (AB).	F.L.P. (AB).	A.L.P. (AB).	Vacuum, inches.	Mean effective pressures. H.P.	I.P.	F.L.P.	A.L.P.	Revolutions.	Indicated horsepower. H.P.	I.P.	F.L.P.	A.L.P.	Total.	Temperatures. Injection.	Discharge.	Feed water. Feed tank.	After passing heater.	Calorimeter. Upper.	Lower.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
2:20	250	83	29.0	8.0	8.0	27.0	...	...	...	...	...	...	...	...	...	...	40	63	51	140	358	310
2:40	252	82	29.0	8.0	7.5	27.0	47.5	15.1	2.9	3.0	112.8	601.6	492.1	177.0	186.4	1,457.1	40	62	50	145	356	300
3:00	250	82	28.5	8.0	7.8	27.0	...	...	...	...	111.8	...	...	...	...	...	39	62	50	155	356	310
3:20	255	81	30.0	8.0	7.5	27.0	48.2	15.6	2.9	2.7	111.9	600.5	500.1	174.1	165.0	1,439.7	40	61	50	150	356	310
3:40	250	84	28.5	8.0	7.5	27.0	...	...	...	...	110.6	...	...	...	...	...	40	64	50	150	358	309
4:00	250	82	27.0	8.0	7 8	27.0	48.1	16.0	2.9	2.7	112.2	601.1	514.6	177.7	165.5	1,458 8	40	63	50	145	358	310
4:20	245	84	27.5	7.8	7.5	27.0	...	...	...	...	111.9	...	...	...	...	...	40	64	50	140	358	308
Total.	1752	579	199.5	55.8	53.5	189.0	143 8	46.7	8.6	8.4	...	1,803.2	1,506.8	528.8	516.9	4,355 6	279	439	351	1,025	2,500	2,157
Avg.	250.3	82.7	28.5	8.0	7.6	27.0	47.9	15.6	2.9	2.8	111.9	601.1	502.3	176 3	172.3	1,451.9	39.9	62.7	50.1	146 4	357 1	308.1
AFTER ENGINE-ROOM OBSERVATIONS.																						
2:20	250	95	30.0	7.5	7.0	27.3	...	...	...	...	...	...	...	...	...	...	44	64	60	...	352	310
2:40	250	95	31.0	7.5	7.5	27.3	48.0	15.5	2.8	3.1	112 8	607.7	504.9	170 8	189.4	1,472.8	44	64	60	...	352	310
3:00	245	95	30.0	7.5	7.5	27.3	...	...	...	...	111.8	...	...	...	...	...	43	63	60	...	352	308
3:20	250	95	32.0	8.0	7.5	27.3	49.7	15 9	2.7	3.3	111.3	617.6	508.3	164.6	198.1	1,488.6	44	64	60	...	352	310
3:40	245	100	32.0	8.0	7.5	27 3	...	...	...	...	110.7	...	...	..	...	...	43	66	60	...	354	310
4:00	248	97	32.0	8 0	7.5	27.2	49.0	16.3	3.1	3.3	112.3	612.9	524.6	190.2	199.4	1,527.1	44	64	60	...	352	309
4:20	245	100	32.0	8.0	8.0	27.2	...	...	...	...	111.9	...	...	...	...	...	44	66	60	...	354	310
Total.	1733	677	219.0	54.5	52.5	190.8	146.7	47.7	8.6	9.6	...	1,838.2	1,537.8	525.6	586.9	4,488 5	306	451	420	...	2,468	2,167
Avg.	247.6	96.7	31.3	7.8	7.5	27.3	48 9	15 9	2.9	3.2	111.79	612.7	512 6	175.2	195.6	1,496.2	43.7	64.4	60	...	352.5	309.6

TABLE D.—*Steam-consumption tests.*—Continued.

TABLE D.—*Steam-consumption tests.*—Continued.

Test No. 5. Date, January 8, 1909.

U. S. S. Birmingham. DATA OF AUXILIARY MACHINERY IN OPERATION.

	Engine-room auxiliaries.																						Fireroom auxiliaries.						Outside auxiliaries.													
	Steam press.			Auxiliary condenser for'd engine room.			Main air pump.				Main circulating pump.				Main feed pumps.							Lift pumps, double strokes.				Blowers.				Flushing pump.		Ice machines.				Dynamo No.—						
					Air and circ. pump.		Double str'kes.		Steam chest press.		Revolutions.		Steam chest press.		Double strokes			Steam chest press.								Revolutions.			Steam-chest press., G.					Starboard.		Port.				Air and circ. pump.			
Time.	Aux. steam line forward, aft, G.	Aux. exhaust line, G.	Vacuum.	Double stroke.	Chest press., G.	Forward pump.	Aft pump.	For'd pump, G.	Aft pump, G.	Forward.	Aft.	Forward, G.	Aft, G.	1.	2.	3.	1 G.	2 G.	3 G.	Forward.	Aft.	Auxiliary.	1 and 2.	3 and 4.	5 and 6.	1 and 2.	3 and 4.	5 and 6.	Double strokes.	Chest pressure, G.	Revolutions.	Chest press., G.	Revolutions.	Chest press., G.	Ampères.	Volts.	Double strokes.	Chest press , G.	Vacuum dynamo cond.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40
2:20	255	5	22	40	60	36	...	55	50	178	130	75	50	...	20	...	...	185	...	5	5.4	...	...	138	248	...	10	35	20	18	120	57	120	62	210	122	24	...	17
	257																							215	282		37	60											
2:40	250	4	23	40	60	36	13.9	55	50	178	130	75	50	...	19	...	...	180	...	6	5.0	...	...	152	248	...	12	35	18	17	120	58	120	62	215	122	24	...	17
	255																							216	282		37	60											
3:00	250	5	22	40	60	36	13.9	55	50	176	128	75	50	...	20	...	...	180	...	6	5.7	...	...	132	256	...	10	35	18	18	120	58	120	61	210	122	24	...	17
	257																							216	282		37	60											
3:20	255	5	22	40	60	36	13.7	55	52	178	128	75	50	...	18	...	...	185	...	8	5.5	...	...	132	260	...	11	35	18	18	120	58	120	61	250	120	24	...	16
	255																							208	286		37	60											
3:40	250	5	24	44	65	36	13.2	55	52	174	128	75	50	...	18	...	...	180	...	4	5.4	...	...	132	256	...	11	35	18	18	120	58	120	62	225	123	24	...	16
	255																							204	282		37	60											
4:00	250	4	24	40	65	36	13.0	55	52	174	128	75	50	...	17	...	...	175	...	6	5.0	...	...	132	236	...	8	35	18	18	120	58	120	62	240	122	24	...	16
	255																							208	282		35	60											
4:20	250	4	22	36	60	36	12.9	55	52	174	126	75	50	...	18	...	...	160	...	7	5.7	...	...	132	244	...	8	36	18	18	120	58	120	62	235	123	24	...	17
	252																							208	264		35	58											
Tot	3,546	32	159	280	430	252	80.4	385	358	1232	896	525	350	...	130	...	...	1245	...	42	37.7	...	...	950	1748	...	70	246	128	125	840	405	848	432	1,585	854	168	...	116
																								1475	1960		255	418											
Avg	253.3	4.6	22.7	40	61.4	36	13.4	55	51.1	176	128	75	50	...	18.6	...	...	178	...	6	5.4	...	...	135.7	250	...	10	35.1	18.3	17.9	120	57 9	121	61.7	226.4	122	24	...	16.6
																								210.7	280		36.4	59.7											
Steam consumption per hour as determined from data of aux. machinery tests.				530		3,426				1,580				2,300						523			1,668						295		1,161				1,610		115		

TABLE D.—*Steam consumption tests.*—Continued.

Test No. 5. Date, January 8, 1909.

U. S. S. Birmingham.—DATA OF CONDENSED EXHAUST STEAM COLLECTED FROM MAIN AND AUXILIARY MACHINERY.

Main engines.						Auxiliaries.								
Forward.			After.											
Interval to fill tank.	Temperature, F.	Weight in tank, in pounds.	Interval to fill tank.	Temperature, F.	Weight in tank, in pounds.	Interval to fill tank.	Temperature, F.	Weight in tank, in pounds.	Interval to fill tank.	Temperature, F.	Weight in tank, in pounds.	Interval to fill tank.	Temperature, F.	Weight in tank, in pounds.
m. s.			*m. s.*			*m. s.*			*m. s.*			*m. s.*		
15:30	50	6,197	14:30	50	6,202	4:45	118	1,139	4:30	128	1,135	4:25	118	1,139
15:30	50	6,172	14:30	51	6,209	4:32	124	1,134	4:15	126	1,130	4:23	120	1,134
15:00	50	6,197	15:00	51	6,202	4:18	124	1,139	4:07	128	1,135	4:16	118	1,139
15:00	50	6,172	13:30	50	6,209	4:19	122	1,134	4:23	130	1,130	4:29	118	1,134
16:00	50	6,197	17:00	48	6,202	4:17	118	1,139	4:28	129	1,135	4:27	120	1,139
15:00	50	6,172	13:15	49	6,209	4:23	120	1,134	4:22	128	1,130	4:24	118	1,134
16:00	50	6,197	14:15	49	6,202	4:23	122	1,139	4:18	128	1,135	4:22	120	1,139
111 in. in tk.	....	5,355	14:30	48	6,209	4:19	126	1,130	4:11	130	1,130	29 in. in tk	120	669
			52½ in. in tk.	....	1,795	4:22	124	1,139	4:09	126	1,135			
						4:28	128	1,130	4:25	120	1,134			
Grand total		48,659	Grand total		51,439	Grand total								31,313
Feed-tank cor't'n		0	Feed-tank cor't'n		—587									
Net water		48,659	Net water		50,852									

TABLE E.—*Steam consumption tests (2d series)—Summary sheet.*

Test No. 6. Date, August 9, 1909. Duration of test, 2 hours 40 minutes. Speed, 14.85 knots.

U. S. S. Salem. STEAM CONSUMPTION PER HOUR, IN POUNDS.

.....	Steam.	Steam.	Steam.	Steam.	I.H.P.	I.H.P.	S.H.P.	Steam per hour.	
								Per I.H.P.	Per S.H.P.
(1) Main turbines, (a) forward	31,826	...	...	...	...	...	1,236.0	...	25.7
(b) after	33,154	...	...	...	...	...	1,270.0	...	26.1
(2) Total for both turbines (by measurement, less amount charged to wet vacuum pumps, which exhaust into main condensers)	...	64,980	...	...	...	...	2,506.0	...	25.9
(3) Auxiliaries essential to the propulsion of the ship (estimated from previous tests), as follows:									
(c) Wet vacuum pumps	876	...	...	...	17.5	...	...	50.0	...
(d) Dry vacuum pumps	2,399	...	...	...	32.5	...	...	73.8	...
(e) Main circulating pumps	1,610	...	...	...	34.4	...	...	46.8	...
(f) Main feed pumps	4,114	...	...	...	49.6	...	...	82.9	...
(g) Forced-draft blowers	2,925	...	...	...	33.6	...	...	87.0	...
(h) Oil pumps	183	...	...	...	.9	...	...	200.0	...
Total for above auxiliaries [(c) + (d) + (e) + (f) + (g) + (h)]	...	12,107	...	...	...	168.5	*168.5	71.8	...
(4) All machinery necessary for propulsion	...	...	77,087	...	...	...	2,674.5	...	28.8
(5) All other purposes, except lift pumps, [(6) − (4)]	...	...	5,196	...	...	...	...	...	...
(6) Total for all purposes, except lift pumps	...	...	...	82,283	...	...	...	...	...
(i) Estimated for lift pumps	...	...	...	664	...	...	...	...	...
(7) Grand total for all purposes (measured)	...	...	...	82,947	...	...	...	...	...

* The I.H.P. of auxiliaries is not converted into S.H.P.

It is assumed in above table that I.H.P. × .94 = S.H.P.

NOTE.—All water and exhaust steam discharged into main condensers is delivered into main measuring tanks. Auxiliary measuring tanks take the remainder. The sum of the two is the total steam used for all purposes.

TABLE E.—*Second steam-consumption tests.*—Continued.

Test No. 6. Date, August 9, 1909.

U. S. S. Salem. FORWARD ENGINE-ROOM OBSERVATIONS.

Time.	Steam pressures. (G) = Gauge. (AB) = Absolute. Number of nozzles open, six (6).											Vacuum, inches.		Shaft data.		Temperatures.					
	In engine room (G).	At H.P. steam chest (AB).	Stages. {Pressures. Temperatures. 1st (AB) temp.	2d (AB) temp.	3d (AB) temp.	4th (AB) temp.	5th (AB) temp.	6th (AB) temp.	7th (AB) temp.	Turbine glands. Frd. (G).	Aft (G).	Spring.	Mercurial.	Starboard. Revs.	Brake H.P.	Injection.	Discharge.	Feed water. Feed tank.	After passing heater.	Calorimeter. Upper.	Lower.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
A.M.																					
8:40	250	177	17.5	10.0	5.0	3.0	1.5	1.1	.8	5	12	28 6	28.8	...	...	53	72	80	180	374	284
			230.0	198.0	158.0	140.0	116.0	100.0	...												
9:00	250	175	19.0	11.0	6.0	3.0	1.5	1.1	.8	5	12	28.6	28.8	208.25	1,266	53	72	71	170	380	300
			230.0	200.0	158.0	140.0	116 0	100.0	...												
9:20	255	195	19.0	10.5	5.5	3.0	1.5	1.1	.8	5	15	28.6	28.8	208.5	1,267	54	72	68	170	380	300
			232.0	200.0	158.0	140.0	116 0	100.0	...												
9:40	250	205	20.0	11.0	6.0	3.3	1.5	1.1	.8	5	15	28.6	28.8	209.5	1,206	52	71	67	170	380	302
			232.0	200.0	160.0	142.0	116.0	98.0	...												
10:00	248	195	19.5	11.5	5.5	3.0	1.5	1.1	.8	5	13	28.6	28.9	209.5	1,206	51	79	68	174	378	302
			230.0	200.0	158.0	140.0	114.0	96.0	...												
10:20	251	190	19.0	10.5	5.5	3.0	1.4	1.0	.9	5	14	28.6	29.0	209.25	1,272	52	78	76	172	376	300
			230.0	198.0	156.0	138.0	112.0	96.0	...												
10.40	250	185	19.0	10.5	5.5	3.0	1.5	1.0	.8	5	14	28.6	29.0	209.0	1,203	52	79	78	172	374	300
			228.0	196.0	156.0	136.0	112.0	94.0	...												
11:00	249	189	19.0	10.5	5.5	3.0	1.5	1.0	.8	5	14	28.6	29 0	208.25	1,199	53	78	78	184	376	302
			228.0	196.0	156.0	138,0	112.0	94.0	...												
11:20	250	185	19.0	10.5	5.5	3.0	1.5	1.0	.8	5	14	28.6	29.0	209.0	1,270	54	79	76	184	380	302
			230.0	198.0	158.0	138.0	114.0	96.0	...												
Tot.	2,253	1,696	171.0	96.0	50.0	27.3	13.4	9.5	7.3	45	123	257.4	260.1	...	...	474	680	662	1,556	3,398	2,692
			2,070.0	1 786.0	1,418.0	1,252.0	1,028.0	874.0	...												
Avg.	250.3	188.4	19.0	10.66	5.55	3.03	1.49	1.05	.81	5	13.7	28.6	28 9	208 9	1,236	52.66	75.5	73.5	172.9	377.5	299.1
			230.0	198.4	157.5	139.1	114.2	97.1	...												

TABLE E.—*Second steam-consumption tests.*—Continued.

Test No. 6. Date, August 9, 1909.

U. S. S. Salem. AFTER ENGINE-ROOM OBSERVATIONS.

Time.	Steam pressures: (G) = Gauge. (AB) = Absolute. Number of nozzles open, 4 (40 minutes), 5 balance of test.											Vacuum, inches.		Shaft data.		Temperatures.					
	In engine room (G).	At H.P. steam chest (AB).	Stages. {Pressures. {Temperatures.							Turbine glands.				Port.				Feed water.		Calorimeter.	
			1st (AB) temp.	2d (AB) temp.	3d (AB) temp.	4th (AB) temp.	5th (AB) temp.	6th (AB) temp.	7th (AB) temp.	Frd. (G).	Aft (G).	String.	Mercurial.	Revs.	Brake H.P.	Injection.	Discharge.	Feed tank.	After passing heater.	Upper.	Lower.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
A.M.																					
8:40	248	233	19.0	10.0	5.0	2.25	1.65	1.05	.85	5.0	4.0	28.9	28.4	...	...	50	74	...	183	392	282
			240.0	201.0	166 0	138.0	115.0	100.0	93.0												
9:00	251	226	20.0	11.0	5.25	2.6	1.65	1.1	.90	5.0	5.5	28.9	28.4	207.8	1,263	52	80	...	176	388	305
			239.0	200.0	169 0	142.0	118.0	103 0	95.0												
9:20	255	231	18.0	10.0	5.0	2.4	1.5	1 05	.80	6 0	4.5	28.9	28.6	209.0	1,270	49	76	...	173	392	304
			241.0	201.0	169.0	142.0	117.0	100.0	90.0												
9:40	249	220	21.0	11.75	6 0	2.7	1.6	1.05	.80	5.5	4 0	28.9	28.7	209.3	1,272	50	78	...	171	385	304
			232.0	199.0	169.0	141.0	118.0	100.0	90.0												
10:00	247	201	20.5	11.25	5.75	2.7	1.6	1.05	.80	5.5	4.0	29.1	28.7	209.3	1,272	50	79	...	171	386	302
			236.0	201.0	169.0	142.0	119.0	100.0	90.0												
10:20	250	215	20.5	11.25	5.3	2.7	1.6	1.05	.80	6.0	3 0	29.1	28.7	209.3	1,272	48	78	...	170	380	305
			239.0	201.0	168.0	141.0	117 0	99.0	90.0												
10:40	249	201	18.0	10.25	5.25	2.5	1.55	1.05	.80	5.5	2.5	29.1	28.7	209 0	1,270	49	78	...	171	378	307
			240.0	202.0	167.0	140 0	115.0	98.0	89.0												
11:00	247	199	19.0	10.25	5.25	2.5	1.55	1.05	.80	7 0	3.5	29 1	28.7	208.5	1,268	49	79	...	179	380	305
			240.0	204.0	167.0	140.0	116.0	98.0	90.0												
11:20	243	215	20.5	11.5	5.25	2 6	1.6	1.05	.80	5.5	3.0	29 1	28.7	208.8	1,270	50	80	...	177	381	305
			237 0	200.0	168.0	140.0	117.0	99 0	90 0												
Tot.	2,239	1,941	176.5	97.25	48.3	22 95	14.3	9.5	7.35	51	34	261.1	257.6	...	...	447	702	...	1,571	3,462	2,719
			2,145.0	1,809.0	1,512.0	1,268.0	1,052.0	897.0	817.0												
Avg.	248.8	215.6	19.5	10.8	5.4	2.6	1.6	1.05	.80	5.7	3.8	29.0	28.6	208.85	1,270	49.7	78	...	174.6	384.7	302.1
			238.2	201.0	168.0	140.9	116.9	99.7	90.8												

TABLE E.—*Second steam-consumption tests.*—Continued.

Test No. 6. Date, August 9, 1909.

U. S. S. Salem. DATA OF AUXILIARY MACHINERY IN OPERATION.

Time.	Engine-room auxiliaries. Steam press. Auxiliary steam line, G.	Steam press. Aux. exhaust line, G.	Aux. cond. for'd eng. room. Vacuum, inches.	Aux. cond. for'd eng. room. Air and circ. pump. Double strokes.	Aux. cond. for'd eng. room. Air and circ. pump. Chest press., G.	Dry vac. pumps, for'd and aft, G. Revolutions.	Dry vac. pumps, for'd and aft, G. Chest pressure	Wet vac. pumps, forward and aft, G Nozzles open.	Wet vac. pumps, forward and aft, G Revolutions.	Wet vac. pumps, forward and aft, G Chest pressure.	Main circ. pumps, for'd and aft, G. Double strokes.	Main circ. pumps, for'd and aft, G. Chest pressure.	Main feed pumps. Double strokes. 1	2	3	Main feed pumps. Steam chest press., G. 1	2	3	Oil pump, forward and aft, G. Double strokes.	Oil pump, forward and aft, G. Chest pressure.*	Fire and bilge pumps, forward and aft, G. Double strokes.	.....	Chest pressure.	Fireroom auxiliaries. Blowers. Revolutions. 1 and 2.	3 and 4.	5 and 6.	Steam chest pressure, G. 1 and 2.	3 and 4.	5 and 6.	Outside auxiliaries. Flushing pump. Double strokes.	Chest pressure, G.	Ice mach., starbd. & port. Revolutions.	Chest pressure, G.	Dynamo No. 3. Amperes.	Volts.	Air & circ. pump. Dbl strokes.	Chest press., G.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38
A.M.																																					
8:40	240					48	70	2	1320.0	90	200	100	....	6	22	....	170	190	52	56	32	....	25	....	256	230	....	55	52			108	58				
	250	7	25	52	90	54	75	2	1243.7	80	240	125	....	6	22	....	170	190	9	125	....	....	....	....	230	232	....	50	65	40	48	84	70	205	125	56	93
9:00	242					48	73	2	1348.8	95	170	100	....	6	25	....	175	190	42	60	32	....	25	....	220	240	....	55	55			108	58				
	255	8	25	54	90	54	75	2	1243.2	80	180	105	....	6	25	....	175	190	8	110	....	....	....	....	212	200	....	50	60	42	48	78	73	210	125	56	95
9:20	250					48	70	2	1353 6	95	190	105	....	6	24	....	185	190	44	60	30	....	20	....	228	240	....	55	60			99	55				
	260	9	25	58	95	54	75	2	1248.0	85	178	105	....	6	24	....	185	190	9	125	....	....	....	....	216	240	....	50	53	40	48	81	72	210	125	58	90
9:40	245					50	75	2	1315.2	100	200	100	....	6	24	....	175	190	44	62	31	....	20	....	232	228	....	55	60			93	55				
	250	8	25	56	90	54	75	2	1248.0	85	168	98	....	6	24	....	175	190	9	125	....	....	....	....	212	216	....	50	53	45	50	84	74	205	125	60	90
10:00	245					49	70	2	1305.0	95	120	65	....	6	22	....	200	190	40	49	30	....	20	....	252	240	....	55	60			102	55				
	250		25	56	90	54	75	2	1243.2	80	145	90	....	6	22	....	200	190	8	120	....	....	....	....	224	216	....	45	53	42	50	84	73	205	125	54	90
10:20	250					50	70	2	1368.0	95	125	65	....	6	23	....	190	190	38	56	30	....	20	....	224	236	....	55	60			108	55				
	255	9	25	56	90	54	75	2	1248.0	85	134	80	....	6	23	....	190	190	10	125	....	....	....	....	208	204	....	45	53	42	50	80	72	240	125	60	95
10:40	252					50	60	2	1368.0	100	120	65	....	6	24	....	190	190	42	49	22	....	20	....	232	236	....	55	55			108	55				
	250	9	25	57	95	54	75	2	1238 4	80	134	80	....	6	24	....	190	190	8	120	....	....	....	....	208	212	....	45	55	42	48	84	77	225	125	60	95
11:00	255					48	60	2	1334 4	95	130	70	....	6	24	....	190	190	44	55	23	....	20	....	220	228	....	50	58			108	55				
	250	9	25	56	90	54	75	2	1243 2	80	136	80	....	6	24	....	190	190	8	120	....	....	....	....	240	240	....	55	53	42	50	72	80	230	125	58	90
11:20	250					48	65	2	1305.6	95	135	75	....	6	23	....	205	190	46	55	24	....	20	....	228	228	....	52	52			108	55				
	250	9	25	56	90	54	75	2	1248.0	85	136	80	....	6	23	....	205	190	10	125	....	....	....	....	240	232	....	53	53	42	48	76	78	215	125	60	93
Tot.	4,489	76	225	501	820	439	613	18	12018 6	860	1390	745	....	54	211	....	1680	1710	392	502	254	....	190	....	2,092	2,106	....	487	512			942	501				
						486	675	18	11203.2	740	1451	843							79	1095				....	1,990	1,992	....	443	498	377	440	723	669	1945	1125	522	831
Avg.	249.4	8.4	25	55.7	91.1	48.8	68.1	2	1335.4	95.5	154.4	82.8	....	6	23.4	...	186 4	190	42.6	55.8	28.2	....	21.1	....	232.4	234.0	....	54.1	55.9	41.9	48.9	104.7	55.7	216.1	125	58	92.3
						54	75	2	1244 8	82.2	161 2	92.6							8.8	121.7					221.1	221.3		49.2	55.3			80.3	74.3				
Steam consumption per hour, as determined from data of auxiliary machinery tests.				428		2,399			876		1,610				4,114				183		226 Lift pump, 664				2,925						752		1,155		1,282		468	

* The upper figures denote oil pressure; the lower figures chest pressure.

machinery tests.

* The upper figures denote oil pressure; the lower figures chest pressure.

TABLE E.—*Second steam-consumption tests.*—Continued.

Test No. 6. Date, August 9, 1909.

U. S. S. Salem.—DATA OF CONDENSED EXHAUST STEAM COLLECTED FROM MAIN AND AUXILIARY MACHINERY.

Main engines.						Auxiliaries.								
Forward.			After.											
Interval to fill tank.	Temperature, F.	Weight in tank, in pounds.	Interval to fill tank.	Temperature, F.	Weight in tank, in pounds.	Interval to fill tank.	Temperature, F.	Weight in tank, in pounds.	Interval to fill tank.	Temperature, F.	Weight in tank, in pounds.	Interval to fill tank.	Temperature, F.	Weight in tank, in pounds
m. s.			*m. s.*			*m. s.*			*m. s.*			*m. s.*		
11:31	78	6,159	11:13	70	6,300	3:40	114	1,110	4:00	108	1,119	4:00	103	1,112
10 52	71	6,215	11:08	66	6,211	3:50	115	1,116	3:40	108	1,110	4:05	107	1,119
12:23	69	6,167	11:12	66	6,200	3:35	118	1,107	3:40	109	1,119	4:10	104	1,112
11:30	67	6,215	11:16	64	6,215	3:35	119	1,116	3:40	106	1,110	3:50	106	1,119
10:55	65	6,167	10:41	63	6,213	3:55	110	1,110	4:00	106	1,119	3:57	107	1,110
10:30	65	6,215	11:11	64	6,215	3:55	108	1,119	4:00	106	1,110	3:58	106	1,119
11:45	64	6,175	11:12	63	6,213	4:00	107	1,110	3:50	109	1,119	4:00	105	1,112
12:40	65	6,215	10:28	63	6,215	4:00	106	1,119	3:50	108	1,110	3:50	106	1,119
11:10	71	6,167	11:15	64	6,213	3:55	106	1,110	4:00	106	1,119	3:55	106	1,110
11:47	73	6,215	12:11	64	6,215	3:45	106	1,119	4:00	106	1,110	4:02	104	1,121
10:04	75	6,159	10:13	64	6,213	3:55	106	1,110	3:55	105	1,121	43 in. in tk.	106	992
12:56	76	6,207	11:18	64	6,215	3:55	106	1,119	4:05	106	1,110			
11:45	76	6,159	12:30	64	6,213	3:55	106	1,110	3:55	105	1,121			
115 in. in tk.	74	5,609	10:57	65	6,211	3:50	108	1,119	4:00	104	1,112			
			65 in. in tk.	66	2,545	3:45	107	1,110	4:15	103	1,121			
Grand total		86,044	Grand total		89,507	Grand total								45,579
Feed tank cor't'n		+ 71	Feed tank cor't'n		0									
Net water		86,115	Net water		89,507									

TABLE F.—*Steam-consumption tests.—Summary sheet.*

Test No. 14. Date, November 22, 1908. Duration of test, 2 hours, 40 minutes. Speed, 15.81 knots.

U. S. S. Chester (4-turbine combination). STEAM CONSUMPTION PER HOUR, IN POUNDS.

.....	Steam.	Steam.	Steam.	Steam.	I.H.P.	I.H.P.	S.H.P.	Steam per hour. Per I.H.P.	Steam per hour. Per S.H.P.
(1) Main turbines, (a) forward	39,780	...	...	...	...	...	1,776.0	...	22.4
(b) after	37,472	...	...	...	...	...	1,837.0	...	20.4
(2) Total for main turbines (by measurement, less amount charged to augmenters, which exhaust into main condensers)	...	77,252	...	...	...	...	3,613.0	...	21.4
(3) Auxiliaries essential to the propulsion of the ship (estimated from previous tests) as follows:									
(c) Augmenters	2,336	...	...	...	46.7	...	...	50.0	...
(d) Main air pumps	4,601	...	...	...	24.7	...	...	186.0	...
(e) Main circulating pumps	979	...	...	...	24.0	...	...	40.8	...
(f) Main feed pumps	5,559	...	...	...	64 0	...	...	86.8	...
(g) Forced-draft blowers	2,297	...	...	...	23 5	...	...	97.7	...
(h) Oil pumps	148	...	...	...	.7	...	...	200.0	...
Total for above auxiliaries [(c) + (d) + (e) + (f) + (g) + (h)].	...	15,920	...	...	...	183.6	*183.6	86.7	...
(4) All machinery necessary for propulsion	...	...	93,172	...	...	...	3,796.6	...	24.5
(5) All other purposes, except lift pumps [(6) − (4)].	...	...	1,708	...	...	...	...	...	...
(6) Total for all purposes, except lift pumps	...	...	...	94,880	...	...	...	...	...
(i) Estimated for lift pumps	...	...	...	741	...	...	...	...	...
(7) Grand total for all purposes (measured)	...	...	...	95,621	...	...	...	...	...

* The I.H.P. of auxiliaries is not converted into S.H.P.

NOTE.—All water and exhaust steam discharged into main condensers is delivered into main measuring tanks. Auxiliary measuring tanks take the remainder. The sum of the two is the total steam used for all purposes.

TABLE F.—*Steam-consumption tests.*—Continued.

Test No. 14. Date, November 22, 1908.

U. S. S. Chester. ENGINE-ROOM OBSERVATIONS.

Time.	Steam pressures. G = Gauge. AB = Absolute.							Vacuum, inches, forward, aft.	Shaft data.									Temperatures.							
						Turbine glands.			Revolutions.				Horsepower.							Feed water, forward and aft.		Calorimeter.				
	In engine room, forward, aft, G.	At H.P. cruising turbine, AB.	At I.P. cruising turbine, AB.	Main H.P. turbine, forward, aft, AB.	L.P. turbine, forward, aft, AB.	For'd engine room, G.	Aft engine room, G.		Starboard. Shaft No. 1.	Starboard. Shaft No. 2.	Port. Shaft No. 3.	Port. Shaft No. 4.	Starboard. Shaft No. 1.	Starboard. Shaft No. 2.	Port. Shaft No. 3.	Port. Shaft No. 4.	Total.	Injection, for'd and aft.	Discharge, for'd and aft.	Air-pump discharge.	After passing heater.	H.P.C., upper, lower.	I.P.C., upper, lower.	Main H.P., upper, lower, forward.	Main H.P., upper, lower, aft.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26
12:00	230	...	...	57	7.5	—.8	...	29.0	...	...	...	...	...	...	...	...	....	52	66	95	219	...	...	316	318
	234			56	8.5	...	0.0	29.6										52	72	97	152			268	267
12:20	230	...	...	57	7.5	2.0	...	29.0	334.1	323.0	328.4	333.8	964.0	905.0	943.2	1009.0	3821.2	51	74	96	230	...	...	320	318
	230			56	8.5	...	2.3	29.6										51	72	98	212			274	275
12:40	230	...	...	57	7.5	—.8	...	29.0	334.7	322.8	329.0	334.9	896.7	904.5	898.8	942.6	3642.6	51	74	100	207	...	...	314	318
	230			56	8.5	...	1.0	29.6										50	70	98	220			278	276
1:00	227	...	...	57	7.5	—.8	...	29.0	335.2	323.7	328 3	334.4	967.2	907.6	942.9	868.8	3686.5	50	74	98	216	...	...	318	320
	232			56	8.5	...	1.0	29.6										50	70	98	230			277	278
1:20	229	...	...	57	7.5	1.0	...	29.0	335.0	323.5	329.2	335.2	828.6	906.4	945.5	873.6	3554.1	50	72	96	227	...	...	318	320
	231			56	8.5	...	1.5	29.6										50	70	98	230			277	278
1:40	229	...	...	57	7.5	.5	...	29 0	335.2	323.6	328.8	335.1	823.3	906.7	944.3	873.3	3547.6	50	73	90	213	...	...	319	321
	231			56	8.5	...	1.5	29.6										52	70	98	224			278	278
2:00	230	...	...	57	7.5	.5	...	29.0	331.6	322.6	329.6	331.6	820.1	903.1	946.6	864.2	3534.0	50	73	97	220	...	...	318	321
	232			56	8.5	...	1.3	29.6										51	70	98	228			277	276
2:20	230	...	...	57	7.5	1.0	...	29.0	334.5	323.4	329.9	334.3	827.3	905.9	947.5	871.2	3551.9	50	73	97	224	...	...	318	321
	228			56	8.5	...	2.0	29.6										51	70	98	226			277	278
2:40	226	...	...	57	7.5	.5	...	29.0	336.2	325.2	330.8	335.4	830.9	911.2	950.0	874.1	3566.2	50	73	98	218	...	...	318	321
	228			56	8.5	...	0.0	29.6										51	72	99	228			277	278
Total.	2061	....	....	513	67.5	3.1	...	261.0	....	....	....	....	....	....	....	....	....	454	652	867	1974	...	...	2859	2878
	2076			504	76.5	...	10.6	266.4										457	636	882	1910			2483	2484
Avg...	229.0	...	...	57	7.5	.34	...	29.0	334.6	323.4	329.3	334.3	869.7	906.3	939.9	897.1	3613.0	50.4	72.4	96.3	219.3	...	...	317.6	319.8
	230.6			56	8.5	...	1.18	29.6										50.8	70.6	98.0	212.2			275.9	276.0

TABLE F.—*Steam-consumption tests.*—Continued.

Test No. 14. Date, November 22, 1908.

U. S. S. Chester. DATA OF AUXILIARY MACHINERY IN OPERATION.

Time.	Engine-room auxiliaries.																						Fireroom auxiliaries.						Outside auxiliaries.									
	Steam press.		Auxiliary condenser for'd engine room	Main air pump.				Main circulating pump.				Main feed pump.				Oil pump.				Lift pump, double strokes.			Blowers.						Flushing pump.		Ice machines.				Dynamo No.			
	Aux. steam line forward, aft, G.	Aux. exhaust line, G.	Vacuum, inches.	Air and circ. pump.		Double str'kes,		Steam chest press.	Revolutions.		Steam chest press.		Double strokes.		Steam chest press.		Double strks.		St'm chest pr's.		Forward.	Aft.	Auxiliary.	Revolutions.			Steam chest press.			Double strokes.	Chest pressure.	Starboard.		P'rt.		Ampères.	Volts.	Air and circ. pump.	
				Double strokes.	Chest press., G.	Forward pump.	Aft pump.	For'd pump, G.	Aft pump, G.	Forward.	Aft.	Forward, G.	Aft, G.	Forward.	Aft.	Forward, G.	Aft, G.	Forward.	Aft.	Forward, G.	Aft, G.	Forward.	Aft.	Auxiliary.	1.	2.	3.	1.	2.	3.	Double strokes.	Chest pressure.	Revolutions.	Chest pressure.	Revolutions.	Chest pressure			Double strokes.	Chest pressure.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41
12 00	234	3.0	25	62	105	26	20	65	80	...	93	44	45	9	20	180	150	...	20	...	40	5	8	48	214	272	...	50	60	...	28	35	78	60	...	...	220	125	42	60
12 20	230	5.0	25	60	90	26	20	65	80	89	94	44	43	24	19	185	150	...	19	...	40	6	16	50	250	274	...	50	60	...	30	33	74	65	...	...	215	125	43	60
11 40	230	2.5	25	60	90	26	20	65	80	87	100	43	43	24	19	185	138	...	19	...	40	6	8	50	250	286	...	50	60	...	29	35	72	57	...	...	225	125	44	60
1 00	233	1.5	25	44	75	26	20	65	80	86	100	42	42	24	19	185	142	...	19	...	37	6	8	46	256	276	...	50	60	...	27	35	72	55	...	...	225	125	42	60
1 20	230	5.0	25	56	65	26	20	65	80	87	100	44	43	20	19	180	145	...	19	...	37	6	8	50	254	266	...	50	60	...	29	35	78	60	...	...	215	125	44	60
1 40	230	7.5	25	46	90	26	20	65	80	85	96	42	43	22	18	180	145	...	19	...	37	6	8	56	252	278	...	50	60	...	29	35	72	57	...	...	215	125	44	60
2 00	230	6.0	25	52	80	26	22	65	80	86	97	43	44	22	18	180	140	...	19	...	37	6	8	48	252	276	...	50	60	...	30	35	70	55	...	...	220	125	44	65
2 20	232	5.5	25	48	75	26	20	65	80	86	96	40	45	20	16	180	140	...	18	...	37	6	8	70	244	274	...	50	60	...	28	35	72	56	...	...	215	125	41	62
2 40	228	8.0	25	30	75	26	20	65	80	85	94	42	45	22	15	180	135	...	17	...	37	5	8	43	214	274	...	45	55	...	29	35	78	53	...	...	220	125	40	65
Tot.	4121	105	225	446	730	232	183	585	720	691	870	384	393	187	163	1,635	1,285	...	169	...	342	52	80	461	2,186	2,476	...	445	535	...	259	313	666	518	...	...	1,970	1125	384	552
Avg.	229	5 8	25	49.5	81.1	25.8	20 3	65	80	86.3	96.6	42.7	43.6	20.8	18.1	181.6	142.7	...	18.7	...	38	5.8	8.9	51.2	242.9	275.1	...	49.4	59.4	...	28 8	34.8	74	57.5	...	...	218.9	125	42.6	61.3
Steam consumption per hour as determined from data of aux. machinery tests.				335		4,601				979				5,559				148				741			2,896						215		345				1,390		245	

TABLE F.—*Steam-consumption tests.*—Continued.

Test No. 14. Date, November 22, 1908.

U. S. S. Chester.—DATA OF CONDENSED EXHAUST STEAM COLLECTED FROM MAIN AND AUXILIARY MACHINERY.

Main engines.						Auxiliaries.								
Forward.			After.											
Interval to fill tank.	Temperature, F.	Weight in tank, in pounds.	Interval to fill tank.	Temperature, F.	Weight in tank, in pounds.	Interval to fill tank.	Temperature F.	Weight in tank, in pounds.	Interval to fill tank.	Temperature, F.	Weight in tank, in pounds.	Interval to fill tank.	Temperature, F.	Weight in tank, in pounds.
m. s.			*m. s.*			*m. s.*			*m. s.*			*m. s.*		
8·22	88	6,155	9·30	90	6,217	2·10	100	1,151	4·35	84	1,155	4·55	86	1,153
9·32	88	6,155	9·10	90	6,136	4·50	100	1,142	3·55	86	1,144	11 in. in tk.	86	252
10·30	94	6,155	9·35	94	6,217	4·15	90	1,153	4·08	88	1,153			
8·35	94	6,157	9·47	94	6,136	4·15	99	1,142	6·27	90	1,144			
9·05	93	6,155	10·43	95	6,217	3·42	82	1,155	6·45	88	1,153			
8·36	95	6,157	9·25	94	6,136	2·43	83	1,146	4·35	90	1,144			
8·41	95	6,157	11·45	94	6,217	4·53	88	1,153	3·46	86	1,153			
8·25	95	6,157	7·15	94	6,136	3·28	90	1,144	4·14	86	1,144			
8·45	93	6,155	10·35	94	3,217	3·45	86	1,153	4·30	84	1,155			
9·03	95	6,157	9·45	94	6,136	5·10	86	1,144	4·25	86	1,144			
10·15	95	6,155	9·55	94	6,217	4·40	82	1,155	5·40	84	1,155			
9·33	95	6,157	9·57	94	6,136	3·35	84	1,146	3·33	86	1,144			
8·07	95	6,155	9·28	94	6,217	3·57	84	1,155	4·52	86	1,153			
8·10	95	6,157	9·33	96	6,136	4·08	88	1,144	4·20	84	1,146			
10·47	95	6,155	9·57	94	6,217	4·40	88	1,153	4·05	82	1,155			
7·50	95	6,157	9·25	96	6,136	4·10	86	1,144	5·30	84	1,146			
9·33	95	6,155	85.5 in. in tk.	96	3,824	4·20	86	1,153	4·00	84	1,155			
100 in. in tk.	95	4,656				3·55	86	1,144	4·25	88	1,144			
Grand total		109,307	Grand total		102,648	Grand total								42,769
Feed tank cor't'n		−115	Feed tank cor't'n		+389									
Net water		109,192	Net water		103,037									

TABLE G.—*Steam consumption tests.*

U. S. S. Birmingham. II.—SYNOPSIS OF HORSEPOWER AND STEAM CONSUMPTION.

Number of test.	Speed per hour in knots.	Horsepower. Machinery necessary to propulsion. Main engines. I. H. P.	Main engines. S. H. P. = .94× I. H. P.	Auxiliaries. I. H. P.	Total. I. H. P. (c+e).	Total. H. P. (d+e).	Condensed exhaust steam (pounds). Per hour. All machinery necessary to propulsion.	All purposes except lift pumps.	All purposes.	Per S. H. P. of main engines. $\frac{h}{d}$	Per H. P. of all machinery necessary to propulsion. $\frac{k}{g}$	Per I. H. P. of main engines. $\frac{h}{c}$	Per I. H. P. for all machinery necessary to propulsion. $\frac{k}{f}$	Per knot. Used by main engines.	Used for all purposes except lift pumps.	Used for all purposes.	Auxiliaries exhaust into—
a	b	c	d	e	f	g	k	l	m	n	o	p	q	r	s	t	u
1	10.03	1,050.2	987.2	52.1	1,102.3	1,039.3	26,689	33,797	34,069	22.3	25.7	20.9	24.2	2,192.2	3,369.5	3,396.7	Engine room auxiliary condenser.
2	10.65	1,236.9	1,162.7	48.0	1,284.9	1,210.7	.....	33,292	33,561	.....	.....	.....	.....	.....	3,126.0	3,151.3	Second receivers main engines.
5	14.95	2,948.1	2,771.2	99.1	3,047.2	2,870.3	49,755	58,729	64,889	65,412	18.0	20.5	16.9	19.3	3,328.1	4,340.4	4,375.4	Engine room auxiliary condenser.
3	15.17	3,118.6	2,931.5	87.0	3,205.6	3,018.5	52,829	60,118	68,510	69,062	18.0	19.9	16.9	18.8	3,482.5	4,516.1	4,552.5	Do.
4	15.35	3,193.7	3,004.0	94.3	3,290.0	3,098.3	.....	.....	66,581	67,118	.....	.....	.....	.....	.....	4,337.5	4,372.5	Second receivers main engines.
8	19.86	7,199.7	6,767.7	208.2	7,407.9	6,975.9	109,820	127,241	130,982	132,037	16.2	18.2	15.3	17.2	5,529.7	6,595.3	6,648.4	Engine room auxiliary condenser.
9	20.00	7,421.8	6,976.5	177.8	7,599.6	7,154.3	.....	.....	131,322	132,380	.....	.....	.....	.....	.....	6,566.1	6,619.0	Second receivers main engines.

6	22.55	11,020.3	10,350.1	307.1	11,327.4	10,666.2	185,210	208,205	213,176	214,894	17.9	19.5	16.8	18.4	8,213.3	9,453.5	9,529.7	Engine room auxiliary condenser.
7	22.80	11,362.0	10,680.3	281.8	11,643.8	10,962.1	.....	.....	213,866	215,891	.....	.....	.....	.....	.....	9,380.0	9,455.7	Second receivers main engines.
10	24.15	14,543.0	13,670.4	528.5	15,071.5	14,198.9	253,372	288,834	294,923	297,300	18.5	20.3	17.4	19.2	10,491.6	12,212.1	12,310.5	Engine room auxiliary condenser.

a Water collected in main measuring tanks unreliable. See remarks at end of test.

TABLE G.—*Steam consumption tests*—Continued.

II.—SYNOPSIS OF HORSEPOWER NECESSARY TO PROPULSION.

Number of test.	Duration of test.	Speed per hour in knots.	Steam pressures (gage).				Vacuum (inches of mercury).		Revolutions per minute.a			Indicated horsepower.								
			Engine room.		Steam chest.							Machinery necessary to propulsion.								
												Main engines.			Auxiliaries.					
			Forward.	Aft.	Forward engine.	After engine.	Forward engine room.	After engine room.	Forward, engine starboard.	After engine, port.	Mean.	Forward, starboard.	After, port.	Both.	Main air pumps.	Main circulating pumps.	Main feed pumps.	Forced draft blowers.	Total (r+s+t+u).	Grand total (q+v).
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q	r	s	t	u	v	w
	H. m.																			
1	4 40	10.03	199.0	195.5	54.2	59.7	25.0	25.8	74.3	74.3	74.3	540.1	510.1	1,050.2	7.5	20.1	19.5	5.0	52.1	1,102.3
2	2 40	10.65	200.0	197.2	55.0	64.4	25.0	25.8	78.9	78.8	78.9	592.7	644.2	1,236.9	6.7	15.0	21.5	4.8	48.0	1,284.9
5	2 00	14.95	250.3	247.6	82.7	96.7	27.0	27.3	111.9	111.8	111.9	1,451.9	1,496.2	2,948.1	18.4	33.9	31.2	15.6	99.1	3,047.2
3	2 00	15.17	248.0	245.3	85.6	96.7	26.0	27.3	113.2	113.2	113.2	1,565.5	1,553.1	3,118.6	10.8	34.0	27.8	14.4	87.0	3,205.6
4	2 00	15.35	250.7	247.9	85.3	97.9	27.0	27.3	114.5	114.5	114.5	1,574.7	1,621.0	3,195.7	16.3	32.8	32.2	13.0	94.3	3,290.0
8	2 40	19.86	248.8	244.0	173.5	184.8	28.2	27.5	150.2	150.0	150.1	3,556.5	3,642.2	7,199.7	27.6	48.0	72.6	60.0	208.2	7,407.9
9	2 00	20.00	249.3	245.4	174.3	185.0	28.0	27.6	151.4	151.4	151.4	3,626.0	3,795.8	7,421.8	20.0	44.0	67.6	46.2	177.8	7,599.6
6	2 40	22.55	250.8	241.1	183.7	201.7	27.7	27.0	173.3	173.3	173.3	5,430.9	5,589.4	11,020.3	28.9	57.6	89.6	131.0	307.1	11,327.4
7	2 00	22.80	234.2	227.1	185.0	202.4	27.9	26.2	175.4	175.4	175.4	5,533.0	5,829.0	11,362.0	32.6	50.0	85.2	114.0	281.8	11,643.8
10	2 40	24.15	238.6	236.4	222.7	236.1	27.5	26.3	192.2	190.8	191.5	7,385.0	7,158.0	14,543.0	27.3	59.6	141.6	300.0	528.5	15,071.5

a As determined from counter readings at beginning and end of test.

TABLE H.—*Steam consumption tests (first series).*

U. S. S. Salem.

II.—SYNOPSIS OF HORSEPOWER AND STEAM CONSUMPTION.

Number of test.	Speed per hour in knots.	Horsepower. Machinery necessary to propulsion.			Condensed exhaust steam (pounds). Per hour.						Per knot.			Auxiliaries exhaust into—
		S. H. P. main turbines.	I. H. P. auxiliaries.	H. P. total (c+d).	Main turbines.	All machinery necessary to propulsion.	All purposes except lift pumps.	All purposes.	Per S. H. P. of main turbines.	Per H. P. of all machinery necessary to propulsion.	Used by main turbines.	Used for all purposes except lift pumps.	Used for all purposes.	
									l / c	g / e	f / b	h / b	k / b	
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q
2	9.8	619	107.1	726.1	32,519	40,296	46,027	46,398	52.5	55.5	3,318.3	4,696.6	4,734.5	Engine room auxiliary condenser.
a 3	10.1	654	106.0	760.0	33,278	40,896	45,731	46,100	50.9	53.8	3,294.8	4,527.8	4,564.3	Do.
4	10.18	711	134.2	845.2	.....	.....	46,644	47,020	.....	.....	.....	4,581.9	4,618.8	Third stage forward turbine.
b 5	10.2	681	124.0	805.0	33,152	41,539	46,225	46,598	48.7	51.6	3,250.2	4,531.7	4,568.4	Engine room auxiliary condenser.
1	10.33	779	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	Do.
6	15.25	2,646	147.3	2,793.3	71,671	82,269	85,945	86,641	27.1	29.4	4,700.0	5,635.7	5,681.4	Do.
7	20.16	7,078	229.8	7,307.8	137,182	152,810	156,259	157,519	19.4	20.9	6,804.6	7,751.0	7,813.4	Do.
9	22.55	10,167	513.9	10,680.9	191,306	220,039	223,028	224,826	18.8	20.6	8,483.6	9,447.0	9,970.1	Do.
8	24.37	14,767	659.4	15,426.4	257,016	292,656	297,176	299,572	17.4	19.0	10,546.4	12,194.3	12,292.6	Do.

a All valves between stages closed.
b All valves between stages open.

TABLE H.—*Steam consumption tests (first series)*—Continued.

II.—SYNOPSIS OF HORSEPOWER NECESSARY TO PROPULSION.

Number of test.	Duration of test.	Speed per hour in knots.	Steam pressure (gage).				Vacuum (inches of mercury).		Revolutions (per minute).[a]			Horsepower.											
			Engine room.		Steam chest.							Machinery necessary to propulsion.											
												S. H. P. main turbines.			I. H. P. auxiliaries.								
			Forward.	After.	Forward.	After.	Forward engine room.	After engine room.	Forward starboard.	After port.	Mean.	Forward starboard.	After port.	Both.	Wet vacuum pumps.	Dry vacuum pumps.	Main circulating pumps.	Main feed pumps.	Forced-draft blowers.	Oil pumps.	Total (r+s+t+u+v+w).	Grand total (q+x).
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q	r	s	t	u	v	w	x	y
	H. m.																					
2	2 00	9.8	218.3	221.8	168.0	195.3	28.4	28.6	133.9	139.4	136.65	296	323	619	17.1	34.2	20.0	21.0	14.0	.85	107.1	726.1
3	2 00	10.1	223.4	222.1	185.0	192.7	28.4	28.6	141.1	140.5	140.8	329	325	654	17.4	34.5	19.6	20.5	13.0	1.00	106.0	760.0
4	2 40	10.18	221.4	221.1	177.2	192.6	28.5	28.7	141.8	142.5	142.1	340	371	711	17.4	40.0	40.1	20.8	15.0	.88	134.2	845.2
5	2 00	10.2	219.8	220.1	178.5	191.7	28.6	28.7	142.2	142.6	142.4	322	359	681	17.2	38.0	32.0	19.9	16.0	.93	124.0	805.0
1	4 00	10.33	218.1	220.2	166.6	192.4	28.9	29.0	144.33	144.09	144.21	375	404	779	.....	.....	.....	.....	.....	.....	.....	.....
6	2 40	15.25	224.4	229.3	193.0	209.4	28.5	28.8	210.9	217.2	214.0	1,174	1,472	2,646	18.2	40.5	29.0	44.6	14.0	1.05	147.3	2,793.3
7	3 00	20.16	252.2	252.4	220.3	207.1	28.6	28.9	284.28	284.27	284.27	3,456	3,622	7,078	23.0	44.5	51.0	56.8	53.4	1.15	229.8	7,307.8
9	2 40	22.55	248.4	247.4	221.2	211.1	28.5	29.0	319.7	319.9	319.8	4,962	5,205	10,167	22.4	42.0	160.0	97.0	191.4	1.13	513.9	10,680.9
8	3 00	24.37	251.1	251.4	241.4	241.1	28.5	28.8	351.5	351.6	351.5	7,376	7,391	14,767	25.8	45.0	197.0	113.4	277.0	1.20	659.4	15,426.4

[a] As determined from counter readings at beginning and end of test.

TABLE I.—*Steam consumption tests (second series).*

U. S. S. Salem.

II.—SYNOPSIS OF HORSEPOWER AND STEAM CONSUMPTION.

Number of test.	Speed per hour in knots.	Horsepower. Machinery necessary to propulsion.			Condensed exhaust steam (pounds). Per hour.						Per knot.			Auxiliaries exhaust into—
		S. H. P. main turbines.	I. H. P. auxiliaries.	H. P. total (c+d).	Main turbines.	All machinery necessary to propulsion.	All purposes except lift pumps.	All purposes.	Per S. H. P. of f / c	Per H. P. of all machinery necessary to propulsion. g / e	Used by main f / b	Used for all purposes except lift pumps. h / b	Used for all purposes. k / b	
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q
1	10.01	719.6	131.66	851.26	30,182	38,636	44,783	45,144	42.0	45.4	3,015.2	4,473.8	4,509.9	Engine room auxiliary condenser.
4	10.06	723.2	122.06	845.26	.....	.....	40,523	40,850	.....	.....	.....	4,028.1	4,060.6	Third stage of forward turbine.
2	12.27	1,477.1	161.38	1,638.48	44,674	54,566	59,610	60,091	30.2	33.3	3,641.0	4,858.2	4,897.4	Engine room auxiliary condenser.
3	12.38	1,474.2	157.69	1,631.89	.....	.....	56,630	57,087	.....	.....	.....	4,574.3	4,611.2	Third stage of forward turbine.
5	12.49	1,487.7	145.98	1,633.68	46,091	55,369	60,246	60,732	30.9	33.9	3,690.0	4,823.5	4,862.4	Engine room auxiliary condenser.
7	14.83	2,463.0	164.11	2,627.11	.....	.....	79,188	79,827	.....	.....	.....	5,339.7	5,382.8	Third stage of forward turbine.
6	14.85	2,505.0	168.51	2,674.51	64,980	77,087	82,283	82,947	25.9	28.8	4,375.7	5,541.0	5,585.6	Engine room auxiliary condenser.
8	17.42	4,720.0	217.03	4,937.03	93,270	107,895	112,603	113,511	19.8	21.9	5,354.1	6,464.0	6,516.1	Do.
10	17.43	4,614.0	191.97	4,805.97	92,325	105,804	110,582	111,474	20.0	22.0	5,297.0	6,344.3	6,395.5	Do.
9	17.44	4,624.0	215.27	4,839.27	.....	.....	107,859	108,729	.....	.....	.....	6,184.6	6,234.4	Third stage of forward turbine.
14	20.05	7,133.0	256.07	7,409.07	131,407	148,385	154,314	155,559	18.4	20.0	6,553.9	7,696.4	7,758.5	Engine room auxiliary condenser.
13	22.47	11,154.0	352.35	11,506.35	182,887	204,691	209,592	211,282	16.4	17.8	8,139.1	9,327.6	9,402.8	Do.
12	23.98	13,954.0	479.29	14,433.29	236,789	266,097	270,239	272,439	17.0	18.4	9,874.4	11,270.0	11,361.0	Do.
11	25.12	18,070.0	681.92	18,751.92	292,751	332,373	338,754	341,486	16.2	17.7	11,654.1	13,485.4	13,594.0	Do.

TABLE I.—*Steam consumption tests (second series)*—Continued.

II.—SYNOPSIS OF HORSEPOWER NECESSARY TO PROPULSION.

Number of test.	Duration of test.	Speed per hour in knots.	Steam pressure (gage).				Vacuum (inches of mercury).		Revolutions (per minute).[a]			Horsepower.											Grand total (q+x).
			Engine room.		Steam chest.							Machinery necessary to propulsion.											
												S. H. P. main turbines.			I. H. P. auxiliaries.								
			Forward.	After.	Forward.	After.	Forward engine room.	After engine room.	Forward starboard.	After port.	Mean.	Forward starboard.	After port.	Both.	Wet vacuum pumps.	Dry vacuum pumps.	Main circulating pumps.	Main feed pumps.	Forced-draft blowers.	Oil pumps.	Total (r+s+t+u+v+w).	
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q	r	s	t	u	v	w	x	y
	H. m.																					
1	2 00	10.01	262.0	261.0	154.7	121.1	28.2	28.5	138.7	140.0	139.3	354.9	364.7	719.6	15.8	34.4	45.6	24.7	10.1	1.06	131.66	851.26
4	2 00	10.06	256.8	254.6	207.0	188.0	28.6	28.6	140.0	140.0	140.0	358.3	364.9	723.2	17.0	33.0	36.0	23.0	12.3	.76	122.06	845.26
2	3 20	12.27	249.8	249.1	215.3	206.2	28.2	28.6	173.5	174.0	173.7	721.4	755.7	1,477.1	16.7	35.0	60.0	27.0	21.8	.88	161.38	1,638.48
3	2 00	12.38	251.9	250.4	217.4	206.6	28.2	28.6	173.5	173.3	173.4	721.4	752.8	1,474.2	16.8	34.4	61.4	27.5	16.8	.79	157.69	1,631.89
5	2 00	12.49	200.3	201.0	30.7	30.0	28.7	28.6	174.9	175.0	175.0	727.5	760.2	1,487.7	17.4	29.4	51.4	25.8	21.2	.78	145.98	1,633.68
7	2 00	14.83	244.8	243.3	171.3	190.8	28.8	28.7	208.7	208.7	208.7	1,200.0	1,263.0	2,463.0	17.2	32.5	30.0	48.5	35.0	.91	164.11	2,627.11
6	2 40	14.85	250.3	248.8	173.4	200.6	28.9	28.6	208.9	208.9	208.9	1,236.0	1,270.0	2,506.0	17.5	32.5	34.4	49.6	33.6	.91	168.51	2,674.51
8	2 40	17.42	245.5	244.0	192.0	207.7	28.7	28.4	246.4	246.5	246.4	2,365.0	2,355.0	4,720.0	21.2	33.5	52.6	63.6	45.2	.93	217.03	4,937.03
10	2 40	17.43	250.0	249.5	192.0	204.0	28.8	28.7	246.4	246.5	246.5	2,276.0	2,338.0	4,614.0	19.3	30.0	42.0	57.6	42.0	1.07	191.97	4,805.97
9	2 00	17.44	250.4	248.4	178.0	197.7	28.7	28.7	246.7	246.7	246.7	2,314.0	2,310.0	4,624.0	23.8	33.5	54.0	60.6	42.4	.97	215.27	4,839.27
14	2 20	20.05	240.0	238.0	187.7	199.4	28.7	28.6	285.1	285.2	285.2	3,485.0	3,668.0	7,153.0	22.4	32.9	71.0	68.5	60.0	1.27	256.07	7,409.07
13	2 00	22.47	246.7	242.4	201.4	200.3	28.6	28.4	322.2	322.2	322.2	5,463.0	5,691.0	11,154.0	23.5	32.0	100.5	85.0	110.0	1.35	352.35	11,506.35
12	2 00	23.98	259.0	253.0	198.0	200.4	28.3	28.1	347.3	347.2	347.2	6,921.0	7,033.0	13,954.0	25.7	33.8	104.0	115.5	198.9	1.39	479.29	14,433.29
11	2 00	25.12	264.7	259.4	249.2	255.7	28.2	28.1	369.9	369.7	369.8	9,080.0	8,990.0	18,070.0	28.7	34.0	135.6	139.2	343.0	1.42	681.92	18,751.92

[a] As determined from counter readings at beginning and end of test.

[b] All nozzles open, speed controlled by throttle.

TABLE J.—*Steam consumption tests (6 turbines).*

U. S. S. Chester.

II.—SYNOPSIS OF HORSEPOWER AND STEAM CONSUMPTION.

Number of test.	Speed per hour in knots.	Horsepower. Machinery necessary to propulsion. S. H. P. main turbines.	I. H. P. auxiliaries.	H. P. total (c+d).	Condensed exhaust steam (pounds). Per hour. Main turbines.	All machinery necessary to propulsion.	All purposes except lift pumps.	All purposes.	Per S. H. P. of main turbines. f/c	Per H. P. of all machinery necessary to propulsion. g/e	Per knot. Used by main turbines. f/b	Used for all purposes except lift pumps. h/b	Used for all purposes. k/b	Auxiliaries exhaust into—
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q
*1	10.03	868.4	118.2	986.6	.....	.....	.....	.....	.....	.....	.....	.....	.....	Engine room auxiliary condenser.
27	10.15	908.0	152.0	1,060.0	25,402	37,641	43,411	43,761	28.1	35.5	2,511.5	4,276.9	4,311.4	Do.
*2	10.65	945.6	121.5	1,067.1	.....	.....	.....	.....	.....	.....	.....	.....	.....	Second stage main H. P. turbines.
*3	12.25	1,758.6	172.4	1,931.0	.....	.....	.....	.....	.....	.....	.....	.....	.....	Engine room auxiliary condenser.
*4	12.75	1,881.7	157.5	2,039.2	.....	.....	.....	.....	.....	.....	.....	.....	.....	Second stage main H. P. turbines.
24	13.00	1,944.9	178.9	2,123.8	41,208	54,993	60,842	61,333	21.2	25.9	3,169.8	4,680.2	4,717.9	Engine room auxiliary condenser.
22	14.83	3,118.4	191.2	3,309.6	55,728	70,249	75,975	76,588	17.9	21.2	3,765.4	5,133.4	5,174.9	Do.
6	15.00	3,131.8	144.2	3,276.0	.....	.....	74,170	74,768	.....	.....	.....	4,944.7	4,984.5	Second stage main H. P. turbines.
*5	15.15	3,131.2	168.6	3,299.8	.....	.....	.....	.....	.....	.....	.....	.....	.....	Engine room auxiliary condenser.
23	15.92	3,919.9	200.2	4,120.1	64,605	79,768	85,204	85,891	16.5	19.4	4,058.1	5,352.0	5,395.2	Do.
26	17.67	5,455.2	253.2	5,708.4	84,968	102,392	111,577	112,477	15.6	17.9	4,809.8	6,314.5	6,365.4	Do.
7	18.03	5,699.2	199.0	5,898.2	94,082	109,279	114,534	115,458	16.5	18.5	5,218.1	6,352.4	6,403.7	Do.
8	18.15	5,977.5	184.8	6,162.3	.....	.....	112,645	113,490	.....	.....	.....	6,206.3	6,252.9	L. P. turbines.

* Steam consumption of this test is not considered reliable.

TABLE J.—*Steam consumption tests (6 turbines)*—Continued.

II.—SYNOPSIS OF HORSEPOWER NECESSARY TO PROPULSION.

Number of test.	Duration of test.	Speed per hour in knots.	Steam pressure (gage).		Vacuum (inches of mercury).		Revolutions (per minute).[a]			Horsepower.										
										Machinery necessary to propulsion.										
										S. H. P. main turbines.			I. H. P. auxiliaries.							
			Forward engine room.	H. P. cruising steam chest.	Forward engine room.	After engine room.	Starboard mean shafts Nos. 1 and 2.	Port mean shafts Nos. 3 and 4.	Mean of all shafts.	Starboard shafts 1+2.	Port shafts 3+4.	All shafts.	Augmentors.	Main air pumps.	Main circulating pumps.	Main feed pumps.	Forced-draft blowers.	Oil pumps.	Total (p+q+r+s+t+u).	Grand total (o+v).
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q	r	s	t	u	v	w
	H. m.																			
1	5 00	10.03	230.8	50.0	28.8	28.8	210.9	205.8	208.3	444.0	424.4	868.4	46.5	14.7	22.2	27.0	7.2	0.62	118.2	986.6
27	2 00	10.15	223.0	50.0	29.0	29.6	207.6	205.1	206.3	437.6	470.4	908.0	46.4	27.5	42.8	31.8	2.5	.96	152.0	1,060.0
2	3 00	10.65	232.7	50.0	28.8	28.3	221.9	212.2	217.1	495.9	449.7	945.6	47.3	17.1	23.8	26.1	6.6	.57	121.5	1,067.1
3	2 40	12.25	229.2	105.2	28.5	29.3	256.8	250.6	253.6	914.0	844.6	1,758.6	46.8	21.1	43.2	48.2	12.4	.72	172.4	1,931.0
4	2 40	12.75	231.9	104.9	28.5	29.3	267.1	260.0	263.6	988.0	893.7	1,881.7	47.0	20.8	37.3	34.3	17.4	.74	157.5	2,039.2
24	2 00	13.00	230.5	105.0	29.0	29.4	271.2	265.6	268.4	1,034.3	910.6	1,944.9	47.8	28.8	54.0	38.5	9.0	.78	178.9	2,123.8
22	2 00	14.80	223.5	150.0	28.8	29.4	310.9	304.8	307.8	1,592.8	1,525.6	3,118.4	45.9	29.0	64.0	40.0	11.5	.78	191.2	3,309.6
6	2 40	15.00	229.3	151.0	28.3	28.9	315.2	309.8	312.5	1,623.9	1,507.9	3,131.8	43.3	15.1	25.6	41.5	18.0	68	144.2	3,276.0
5	2 40	15.15	232.0	151.0	28.3	29.0	318.6	313.6	316.1	1,611.2	1,520.0	3,131.2	47.4	18.6	31.2	53.3	17.4	.74	168.6	3,299.8
23	2 00	15.92	226.8	180.0	28.8	29.4	335.7	329.5	332.6	2,031.2	1,888.7	3,919.9	47.2	30.3	66.8	41.5	13.6	.81	200.2	4,120.1
26	2 00	17.67	243.0	235.0	29.0	29.6	372.9	369.6	371.2	2,761.5	2,693.7	5,455.2	50.8	31.2	101.0	51.0	18.0	1.17	253.2	5,708.4
7	2 40	18.03	243.9	235.0	28.5	29.4	380.5	377.5	379.0	2,909.5	2,789.7	5,699.2	50.6	16.3	40.0	72.6	18.0	1.49	199.0	5,898.2
8	2 00	18.15	247.5	235.0	28.5	29.4	383.1	379.5	381.3	3,016.0	2,961.5	5,977.5	51.4	14.3	32.0	64.5	21.0	1.59	184.8	6,162.3

[a] As determined from counter readings at beginning and end of test.

TABLE K.—*Steam consumption tests (5 turbines).*

II.—SYNOPSIS OF HORSEPOWER AND STEAM CONSUMPTION.

Number of test.	Speed per hour in knots.	Horsepower. Machinery necessary to propulsion.			Condensed exhaust steam (pounds). Per hour.						Per knot.			Auxiliaries exhaust into—
		S. H. P. main turbines.	I. H. P. auxiliaries.	H. P. total (c+d).	Main turbines.	All machinery necessary to propulsion.	All purposes except lift pumps.	All purposes.	Per S. H. P. of main turbines.	Per H. P. of all machinery necessary to propulsion.	Used by main turbines.	Used for all purposes except lift pumps.	Used for all purposes.	
									$\frac{f}{c}$	$\frac{g}{e}$	$\frac{f}{b}$	$\frac{h}{b}$	$\frac{k}{b}$	
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q
9	10.43	828.5	126.8	955.3	33,927	44,347	47,993	48,380	40.9	46.4	3,252.8	4,601.4	4,638.5	Engine room auxiliary condenser.
10	10.56	826.4	125.1	951.5	.....	.....	46,164	46,522	.....	.....	.....	4,371.6	4,405.5	Second stage main high-pressure turbines.
12	15.10	3,053.6	154.2	3,207.8	63,565	75,818	80,815	81,467	20.8	23.5	4,209.6	5,352.0	5,395.2	Engine room auxiliary condenser.
13	15.35	3,166.5	146.9	3,316.4	.....	.....	77,600	78,286	.....	.....	.....	5,059.3	5,100.0	Low-pressure turbines.
16	20.00	8,154.3	245.7	8,400.0	119,300	137,376	140,100	141,230	14.6	16.4	5,968.0	7,005.0	7,061.0	Engine room auxiliary condenser.
17	20.14	8,368.8	245.1	8,613.9	.....	.....	136,649	137,751	.....	.....	.....	6,785.0	6,839.6	Low-pressure turbines.
20	22.62	12,467.8	309.8	12,777.6	159,435	181,336	186,430	187,933	12.8	14.2	7,048.4	8,241.8	8,308.3	Engine room auxiliary condenser.

TABLE K.—*Steam consumption tests (5 turbines)*—Continued.

II.—SYNOPSIS OF HORSEPOWER NECESSARY TO PROPULSION.

Number of test.	Duration of test.	Speed per hour in knots.	Steam pressure (gage).		Vacuum (inches of mercury).		Revolutions (per minute).[a]			Horsepower.										
										Machinery necessary to propulsion.										
										S. H. P. main turbines.			I. H. P. auxiliaries.							
			Forward engine room.	I. H. P. cruising turbines.	Forward engine room.	After engine room.	Starboard mean shafts Nos. 1 and 2.	Port mean shafts Nos. 3 and 4.	Mean of all shafts.	Starboard shafts, 1+2.	Port shafts, 3+4.	All shafts (m+n).	Augmentors.	Main air pumps.	Main circulating pumps.	Main feed pumps.	Forced draft blowers.	Oil pumps.	Total (p+q+r+s+t+u).	Grand total (o+v).
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q	r	s	t	u	v	w
	H. m.																			
9	2 40	10.43	230.1	15.0	29.2	29.5	206.7	217.7	212.2	415.0	413.5	828.5	46.8	18.1	20.1	32.3	9.0	.51	126.8	955.3
10	2 40	10.56	232.0	15.0	29.3	28.8	213.2	217.2	215.2	412.3	414.1	826.4	47.1	17.9	18.0	31.0	10.5	.61	125.1	951.5
12	3 00	15.10	229.3	60.0	29.3	29.3	306.4	322.5	314.5	1,319.4	1,734.2	3,053.6	46.7	22.2	38.0	36.5	10.0	.81	154.2	3,207.8
13	2 20	15.35	222.7	60.0	29.3	29.7	311.1	329.3	320.2	1,336.9	1,832.6	3,169.5	44.7	22.6	30.0	37.4	11.5	.74	146.9	3,316.4
16	2 40	20.00	228.0	145.0	29.0	29.6	408.3	435.5	421.9	3,378.1	4,776.2	8,154.3	46.9	29.8	87.0	71.0	10.0	.98	245.7	8,400.0
17	2 40	20.14	228.4	145.0	29.0	29.6	411.8	438.5	425.2	3,529.6	4,839.2	8,368.8	47.2	29.7	82.0	76.8	8.5	.86	245.1	8,613.9
20	2 40	22.62	230.4	210.0	28.5	29.4	465.9	496.9	481.4	5,404.7	7,065.1	12,467.8	48.5	31.1	110.0	92.0	27.0	1.16	309.8	12,777.6

[a] As determined from counter readings at beginning and end of test.

TABLE L.—*Steam consumption tests (4 turbines).*

II.—SYNOPSIS OF HORSEPOWER AND STEAM CONSUMPTION.

Number of test.	Speed per hour in knots.	Horsepower.			Condensed exhaust steam (pounds).									Auxiliaries exhaust into—
		Machinery necessary to propulsion.			Per hour.						Per knot.			
		S. H. P. main turbines.	I. H. P. auxiliaries.	H. P. total (c+d).	Main turbines.	All machinery necessary to propulsion.	All purposes except lift pumps.	All purposes.	Per S. H. P. of main turbines. $\frac{f}{c}$	Per H. P. of all machinery necessary to propulsion. $\frac{g}{e}$	Used by main turbines. $\frac{f}{b}$	Used for all purposes except lift pumps. $\frac{h}{b}$	Used for all purposes. $\frac{k}{b}$	
a.	b	c	d	e	f	g	h	k	l	m	n	o	p	q
11	10.15	746.6	143.6	890.2	33,539	45,544	47,291	47,690	44.9	51.2	3,304.3	4,659.2	4,698.5	Engine room auxiliary condenser.
15	15.68	3,463.8	180.3	3,644.1	.....	.....	93,864	94,621	.....	.....	.....	5,986.2	6,034.5	L. P. turbines.
14	15.81	3,613.0	183.6	3,796.6	77,252	93,172	94,880	95,621	21.4	24.5	4,886.3	6,001.2	6,048.1	Engine room auxiliary condenser.
18	20.05	7,834.7	284.8	8,119.5	132,651	151,585	156,954	158,220	16.9	18.7	6,616.0	7,828.0	7,891.3	Do.
19	20.05	7,849.5	297.3	8,146.8	.....	.....	154,749	155,997	.....	.....	.....	7,718.0	7,780.9	L. P. turbines.
21	22.78	12,156.4	338.8	12,495.2	176,551	198,803	203,909	205,553	14.5	15.9	7,750.3	8,951.2	9,023.4	Engine room auxiliary condenser.
29	23.85	15,651.0	545.9	16,196.9	215,117	247,473	252,164	254,198	13.7	15.3	9,019.6	10,572.9	10,658.2	Do.
25	23.92	15,185.8	385.9	15,571.7	221,173	246,907	254,783	256,838	14.6	15.9	9,246.3	10,651.5	10,737.4	Do.
28	24.67	18,544.0	606.9	19,150.9	253,662	290,861	293,687	296,054	13.7	15.2	10,282.2	11,904.6	12,000.6	Do.

TABLE L.—*Steam consumption tests (4 turbines)*—Continued.

II.—SYNOPSIS OF HORSEPOWER NECESSARY TO PROPULSION.

Number of test.	Duration of test.	Speed per hour in knots.	Steam pressure (gage).				Vacuum (inches of mercury).		Revolutions (per minute).[a]			Horsepower.										
			Engine room.		Steam chest.							Machinery necessary to propulsion.										
												S. H. P. main turbines.			I. H. P. auxiliaries.							
			Forward.	After.	Forward main H. P.	After main H. P.	Forward engine room.	After engine room.	Starboard mean shafts Nos. 1 and 2.	Port mean shafts Nos. 3 and 4.	Mean of all shafts.	Starboard shafts 1+2.	Port shafts 3+4.	All shafts (o+p).	Augmenters.	Main air pumps.	Main circulating pumps.	Main feed pumps.	Forced draft blowers.	Oil pumps.	Total (r+s+t+u+v+w+x).	Grand total (z+y).
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q	r	s	t	u	v	w	x	y
	H. m.																					
11	2 40	10.15	226.8	231.5	6.0	6	29.1	29.4	205.9	207.8	206.8	402.4	344.2	746.6	46.3	18.3	23.0	46.5	9.0	0.55	143.6	890.2
15	2 00	15.68	225.8	231.2	42.0	41	28.6	29.6	325.1	329.0	327.1	1,716.8	1,747.0	3,463.8	46.6	25.2	22.5	60.2	25.0	.78	180.3	3,644.1
14	2 40	15.81	229.0	230.6	42.0	41	29.0	29.6	329.0	331.8	330.4	1,776.0	1,837.0	3,613.0	46.7	24.7	24.0	64.0	23.5	.74	183.6	3,796.6
18	2 40	20.05	217,7	220.7	87.5	87	28.8	29.6	422.3	421.8	422.1	3,939.7	3,895.0	7,834 7	44.6	26.7	123.0	79.0	10.5	.98	284.8	8,119.5
19	2 20	20.05	219.6	224.7	87.5	87	28.8	29.5	422.6	421.9	422.3	3,931.9	3,917.6	7,849.5	45.5	28 8	130.0	82.0	10.0	1.04	297.3	8,146.8
21	2 40	22.78	222.2	228.9	126.0	126	28.5	29.2	484.4	481.2	482.8	6,068.5	6,067.9	12,136.4	46.1	27.7	136.0	90.0	37.5	1.52	338.8	12,495.2
29	1 00	23.85	220.0	232.75	156.0	156	28.0	29.0	513.9	509.1	511.5	7,849.7	7,801.3	15,651.0	47 1	45.2	246.0	118.8	87.0	1.79	545.9	16,196.9
25	1 00	23.92	224.5	230.5	156.0	156	28.1	28.9	515.5	512.6	514.1	7,768.8	7,417.0	15,185.8	47.3	28.9	118.0	111.6	78.9	1.22	385.9	15,571.7
28	2 00	24.67	228.3	241.3	185.0	185	28.0	28.8	544.8	540.2	542.5	9,365.0	9,179 0	18,544.0	48.1	57.7	276.0	121.5	101.5	2.14	606.9	19,150.9

[a] As determined from counter readings at beginning and end of test.

III. Coal-Consumption Tests at Speeds of Approximately 10, 15 and 20 Knots.

IV. Coal-Consumption Test at Full Power.

Four comparative coal-consumption tests were carried out as outlined above, the vessels being in company, and consequently subjected to the same weather conditions. As planned, these tests were to be of 100, 50, 100 and 24 hours' duration, respectively, but on account of rough weather the 10-knot trial was discontinued after steaming 96 hours, and for a similar reason the 20-knot trial ended after 98 hours.

Bradford, R. I., was selected as an accessible and suitable base from which to operate, particularly as the same grade of coal (New River) was available at this coaling station for all trials. The use of the same grade of coal, it may be pointed out, is a desirable, if not a necessary, condition to accurate comparison.

Before each coaling an estimate was made for each vessel of the amount necessary to be taken on board, so that the mean displacement for each trial, as nearly as could be figured beforehand, would average 4,000 tons. This calculation included, in addition, an allowance for "make-up" feed water, which in all coal-consumption trials was carried in double-bottom compartments, the distilling plant being operated only to supply fresh water for purposes other than the boilers. The displacement, as finally determined, was figured from draughts taken at the beginning and end of each trial, corrected for weights removed while reaching the course, and in steaming to an enchorage after completion of the trial.

On coal-consumption tests the aim was to effect a comparison of the sea performance of the three vessels as measured by the coal consumed. As outlined originally, shifting of an engineers' personnel (firerooms only) from ship to ship was contemplated, with a view of thus fixing with precision the number in the complement, and, in addition, eliminating as far as possible the personal factor in firing. This plan, however, was abandoned because of practical difficulties in

successfully carrying out the scheme, and for the further reason that, under such circumstances, the various trials of all vessels would have had to be separately carried out, instead of steaming in company, with resulting complications as to influence of weather conditions. All trials were run, therefore, with the engineers' force regularly assigned to the vessels, the complements of which, with reference to numbers in each rating, differed somewhat; but it is impossible under the circumstances, to frame conclusions with accuracy covering the comparative efficiencies of the three engineering personnels.

No attempt was made to directly measure the steam used by either the main or auxiliary machinery, the water-measuring apparatus having been removed from all ships prior to commencement of these trials. Attention was concentrated primarily on securing an accurate determination of the quantity of coal consumed, although on all trials such detailed data were recorded as would enable a close calculation to be made of the steam used by any or all of the machinery in operation by comparison with previous steam-consumption tests.

As accuracy of comparison on trials of this character depends almost exclusively on the accuracy with which the quantity of coal burned is ascertained, the method of securing such data received careful attention when the trials were planned. It is obvious that any practical method to obtain such information adaptable to conditions on shipboard is apt to involve an element of doubt, and, therefore, the method selected should be the one least likely to error. The following were suggested and considered: First, to weigh all coal as taken from the bunkers; second, to use only coal put up in bags of known weight; third, to tally buckets or baskets of coal as taken from the bunkers, and at intervals to ascertain average weight; fourth, to mark the coal bunkers horizontally at intervals of 100 cubic feet with painted stripes, so that by leveling at the beginning and end of each trial the number of cubic feet of coal taken from each bunker could be calcu-

lated. Any of the four methods mentioned involve uncertainties as to correct weight, the most important reasons therefor being as follows: (1) The weighing of coal as taken from bunkers on account of the crowded condition of fire-rooms would hardly be a feasible scheme on the scout cruisers at high powers, no matter how many extra men might be assigned for the purpose; (2) it would have been expensive to bag coal in sufficient quantities for the proposed trials, and, furthermore, bagging material ordinarily used generally becomes injured in handling on shipboard, resulting in coal being spilled and for which it is difficult to make accurate allowance; (3) experience shows that calculations for coal expenditures by tallying buckets, even when great care is exercised, almost invariably fall short of the true weight, undoubtedly attributable to the tendency on part of the fire-room staff to "heap" buckets, in addition to the constant liability of error (generally one way only) in tallying, which latter, it may be noted, is a necessary adjunct to the first two methods; (4) in stowing of coal bunkers, there is never absolute certainty that all pockets or spaces have been properly filled, especially around beams, stiffeners, platforms, etc., and, moreover, the weight of coal which can be stored in a given bunker is governed to some extent by the relative proportions of slack and lump.

The fourth method was selected as being open to the fewest objections and as offering the best practical means of comparing results of coal-consumption trials. In this connection it may be stated that on preliminary government-acceptance trials of these vessels (for 24-hour trials at 12 and 22½ knots, respectively) it was the intention to make use of this method of determining coal consumption and the bunkers consequently were striped originally at the building yards. On the trials referred to, however, the *Chester* was the only ship on which the method was used.

On all trials, comparison is based upon number of cubic feet of coal consumed. In order, however, to convert such results into units of weight, the cubical space contained in

cars used in transporting coal from storehouses to the wharf when coaling was calculated and corresponding weights of coal ascertained. The cars used were small and approximately of two and four tons capacity. They had a plane side surface, the base being constructed with a triangular jog in the central part and extending the full length to facilitate emptying. As coaling proceeded, preliminary to each test, a number of cars in use were measured; each was weighed when empty, and again when loaded and leveled. The average of all calculations showed a ton of coal to occupy a space of 43.5 cubic feet, and although this figure seems somewhat high, it has been adopted in computing coal consumed in tons on all trials.

After coaling and prior to commencement of each trial, such bunkers (after leveling) as were to be used, were carefully inspected, and the distance noted from top of coal to the nearest stripe, from which data cubical contents were readily calculated. While the distance between adjacent stripes varied in the same bunker, and, of course, for different bunkers, the average was approximately seven inches for those bunkers abreast the firerooms. When leveled, readings by independent observers rarely differed for any bunker by as much as fifteen cubic feet, and this figure may be regarded as representing the maximum liability to personal error, assuming the stripes to have been properly located and the coal closely stowed. It should be pointed out here that the tendency of bunker coal is to settle when ships roll at sea, which tendency is pronounced at times, and especially in bunkers indifferently stowed, where there has been failure to fill in properly around obstructions (girders, beams, platforms, etc.) when coaling. Directions were given, however, that particular care be exercised in this respect in stowing bunker coal for these trials.

In general, the manner of securing coal data on the various trials was as follows: In steaming from the anchorage off Bradford to the open sea, and prior to commencement of each trial, coal was taken from bunkers other than those reserved for the trial. At the instant of starting, fireroom plates were swept clean, or an estimate made of the coal thereon, and

doors raised on those bunkers from which coal was to be used. Upon completion of the trial, these bunker doors were lowered, care having been taken to leave no coal, or as little as possible, on the floor plates at that time. The coal remaining in the trial bunkers was then leveled and inspected, and the volume figured from the stripes, coal for the boilers in operation as the ship proceeded to port being taken from other bunkers. On the 20-knot trial a slight modification was required, since it was necessary during this trial to make use of coal stowed in a larger number of bunkers, and in consequence those bunkers from which coal was taken until the trial commenced were releveled and inspected.

During each trial a sample of coal was taken on the three vessels, and later subjected to analysis, as shown below:

Coal consumption tests

COAL ANALYSIS.

Ship.	No. of test.	Proximate analysis:										
		Coal as fired.						Dry coal.				
		Moisture.	Volatile matter.	Fixed carbon.	Ash.	Sulphur	B T U's.	Volatile matter	Fixed carbon.	Ash.	Sulphur.	B. T. U.'s.
Birmingham	1	1.39	20.91	70.80	6.90	0.86	14,311	21.20	71.80	7.00	0.87	14,513
	2	1.37	21.06	72.36	5.21	.90	14,559	21.35	73.37	5.28	.91	14,761
	3	1.27	20.79	72.22	5.72	.75	14,478	21.06	73.15	5.79	.76	14,664
	4	1.49	21.29	71.08	6.14	.69	14,409	21.61	72.16	6.23	.70	14,627
Salem	1	1.37	21.43	72.17	5.03	.77	14,598	21.73	73.17	5.10	.78	14,801
	2	2.43	21.02	66.76	9.79	.66	13,604	21.54	68.42	10.04	.67	13,943
	3	1.49	20.24	72.45	5.82	.73	14,417	20.55	73.55	5.90	.74	14,635
	4	1.53	20.50	71.56	6.41	1.07	14,390	20.82	72.67	6.51	1.08	14,614
Chester	1	1.27	20.65	72.81	5.27	.81	14,481	20.92	73.74	5.34	.82	14,667
	2	3.34	19.30	69.86	7.60	.71	13,958	19.95	72.20	7.85	.73	14,425
	3	1.42	20.10	73.51	4.97	.85	14,548	20.39	74.57	5.04	.86	14,758
	4	1.04	20.92	73.34	4.70	.74	14,716	21.14	74.11	4.75	.75	14,871
Mean		1.61	20.68	71.58	6.13	.80	14,372	21.02	72.74	6.24	81	14,607

During the trials at 10, 15 and 20 knots each vessel regulated revolutions of main propelling machinery for the respective speeds mentioned in accordance with standardization curves (Plates 8–10). Generally a column-like formation was preserved, although there was no attempt to maintain position with exactness. Revolutions were varied occasionally, but

only as was necessary to keep the vessels within a reasonable distance of one another. Such changes were always slight, and usually made at long-time intervals, so that no ship was handicapped—as often happens when steaming regularly in squadron—by constant throttle-valve changes to regulate speed. A number of turns were made during the 10 and 15-knot trials; on the 20-knot run but one turn was made. It should be mentioned that at each turn the leading vessel took up position astern in the column, and that at such times navigators of the three vessels established their relative positions by stadimeter of observations, which was also done at the commencement and end of each trial. Distances covered by each vessel on the various trials, as figured from revolutions, when taken in conjunction with observed stadimeter data, indicates a considerable influence on speed, apparently attributable to some, or all, of the following: Trim of vessels; steering; weather conditions.

On full-power run, revolutions were not set, all vessels being permitted to proceed on a course previously laid out at the highest speed possible to maintain. A number of 180 degrees turns were made, in which way all vessels were brought within reasonable distances of one another when too widely separated. Soon after this trial began, the forward propelling engine of the *Birmingham* developed a knock in wrist-pin brasses of I.P. crosshead which increased gradually and finally resulted in carrying away the oiling gear to this part of the machinery. In addition, a leak developed in receiver piping of the same engine—connection from H.P. to I.P. valve chests at the top—but no serious inconvenience resulted therefrom, except increase of "make-up" feed water required as the trial progressed. Because of these mishaps, it was believed inadvisable to extend the trial of this vessel beyond twelve hours, and at the end of that time, the *Birmingham* returned to port. The other vessels, however, continued on the trial for the full period of twenty-four hours.

Plate 13 (pounds of coal per hour—speed in knots) has been laid down from results of coal-consumption trials; a

TABLE M.—*Coal-consumption tests.*

No. of test.	Approximate speed.	Duration of test in hours.	Birmingham.				Salem.				Chester.			
			Average pressure in fireroom in inches of water.											
			Number of boilers in use.	No. of fireroom.			Number of boilers in use.	No. of fireroom.			Number of boilers in use.	No. of fireroom.		
				1	2	3		1	2	3		1	2	3
1	10	96	3	.....	.....	0.694	4	.....	0.325	.....	4	.....	.....	.....
2	15	50	8	.....	0.546	.788	8	.....	.407	0.418	7	.....	.....	.....
3	20	98	12	1.227	1.214	1.174	12	1.57	1.58	1.554	12	.....	.....	[a]0.9
4	(b)	[c]24	12	5.077	4.942	5.807	12	5.08	4.84	5.11	12	3.03	3.03	3.03

TOTAL EVAPORATION PER HOUR—BLOWER STEAM—NET AVAILABLE STEAM.

			Total evaporation.	Blower steam.	Net available steam.	Total evaporation.	Blower steam.	Net available steam.	Total evaporation.	Blower steam.	Net available steam.
			A[d]	B[e]	(A-B)[f]	C[d]	D[e]	(C-D)[f]	E[d]	F[e]	(E-F)[f]
1	10	96	32,871	990	31,881	44,674	873	43,801	36,097	.....	36,097
2	15	50	67,438	1,961	65,477	84,280	2,219	82,061	74,596	.....	74,596
3	20	98	129,176	5,200	123,976	163,111	7,385	155,726	134,270	525	133,745
4	(b)	[c]24	271,545	18,493	253,052	293,103	24,820	268,283	314,161	6,510	307,651

PER CENT OF TOTAL EVAPORATION PER HOUR—BLOWER STEAM—NET AVAILABLE STEAM.

1	10	96	100	3	97	100	2	98	100	0	100
2	15	50	100	3	97	100	3	97	100	0	100
3	20	98	100	4	96	100	4	96	100	0.4	99.6
4	(b)	[c]24	100	7	93	100	8	92	100	2	98

[a] Forced draft used intermittently 50 hours.
[b] Maximum speed: Birmingham, 24 knots; Salem, 24.32 knots; Chester 25.08 knots.
[c] Birmingham discontinued this trial after twelve hours.
[d] Total water evaporated per hour, in pounds. (II. Steam consumption tests.)
[e] Steam used per hour by forced draft blowers, in pounds. (See auxiliary tests.)
[f] Net available steam per hour, in pounds.

TABLE N.—*U. S. Scouts Birmingham, Salem, and Chester coal consumption tests.*

III-IV.—SYNOPSIS OF HORSE POWER, STEAM AND COAL CONSUMPTION.

Number of test.	Duration of test.	Name of ship.	Displacement.	Speed per hour in knots.[a]	Total distance run.	Horsepower.					Steam.					Coal for all purposes per—						Auxiliaries exhaust into—
						Machinery necessary to propulsion.					For all purposes per hour from curves of steam consumption tests.	Make up feed.[a]		Actual total on test in pounds (n+p).	Reduced to water evaporated from and at 212° per pound of coal.	Day.		Hour.				
						Main engines or turbines.		Auxiliaries.	Total.			Tons per day.	Pounds per hour.		Actual on this test.	As established by boiler evaporative tests.	Cubic feet.	Tons (43.5 cubic feet=1 ton).	Pounds.	Pounds per hour per knot.	Pounds per square feet of grate surface.	Per I. H. P. of machinery necessary to propulsion.	Per H. P. of machinery necessary to propulsion.	
						I. H. P.	S. H. P.=.94×I. H. P.	I. H. P.	I. H. P. (g+k).	H. P. (h+k).													
a	b	c	d	e	f	g	h	k	l	m	n[c]	o	p	q	r	s	t	u	v	w	x	y	z	
[d]1	*Hrs* 90	Birmingham	3,994.0	9.86	946.56	960	902.4	61.4	1,021.4	963.8	32,871	10.10	942.2	33.813	11.60	11.12	1,398	32.14	2,999	304	17.02	2.94	3.11	Second receiver, both engines.
		Salem	4,022.6	9.90	950.40	.....	668.4	115.1	.....	783.5	44,674	11.66	1,088.3	45.762	9.52	10.85	2,342	53.85	5,026	508	21.67	.....	6.41	E.R.auxiliary condenser, 40 minutes; main condenser, 20 hours 40 minutes; remainder third stage, forward turbine.
		Chester	4,026.9	10.03	962.88	.....	730.0	105.1	.....	835.1	36,097	10.97	1,023.8	37,121	9.54	9.64	1,830	42.06	3,926	391	16.92	.....	4.70	Port L. P. turbine.

[e]2	50	Birmingham	4,001.9	15.00	750.00	3,263	3,067.2	89.2	3,352.2	3,156.4	67,438	13.90	1,297.3	68,735	10.87	11.20	3,098	71.21	6,646	443	14.32	1.98	2.105	Second receiver, both engines.
		Salem	3,964.5	14.91	745.50	.....	2,609.2	170.2	.....	2,779.2	84,280	12.12	1,131.2	85,411	9.06	10.86	4,665	107.23	10,008	671	21.56	....	3.00	Third stage, forward main turbine.
		Chester	3,967.8	14.98	749.00	.....	3,201.3	128.1	.....	3,329.4	74,596	13.20	1,232.0	75,828	9.73	9.50	3,887	89.36	8,341	557	20.54	....	2.51	Port L. P. turbine.
[f]3	98	Birmingham	3,960.0	19.83	1,943.34	7,575	7,120.5	188.7	7,763.7	7,309.2	129,176	26.10	2,436.0	131,612	9.62	10.90	6,675	153.45	14,322	722	20.58	1.84	1.96	Second receiver, both engines.
		Salem	4,030.5	20.25	1,984.50	.....	6,883.3	329.0	.....	7,212.3	163,111	17.51	1,634.2	164,745	9.54	10.60	8,788	202.03	18,855	931	27.09	....	2.61	Third stage, forward main turbine.
		Chester	3,970.3	19.90	1,950.20	.....	8,374.0	206.2	.....	8,580.2	134,270	16.80	1,568.0	135,838	9.55	9.44	7,086	162.90	15,205	760	21.84	....	1.77	Starboard L. P. turbine.
[g]4	12	Birmingham	4,059.0	24.00	288.00	14,828	13,938.4	535.9	15,363.9	14,474.3	271,545	120.4	11,237.3	282,782	8.42	9.52	16,344	375.72	35,070	1,461	50.38	2.28	2.42	Main condensers.
	24	Salem	4,028.0	24.32	583.68	.....	14,630.0	695.8	.....	15,325.8	293,103	45.63	4,258.3	297,361	8.36	9.25	18,060	415.15	38,749	1,593	55.74	....	2.53	Main condenser, 30 minutes; remainder fourth stage, main turbines.
	24	Chester	3,961.6	25.08	601.92	.....	19,632.0	463.6	.....	20,095.6	314,161	27.00	2,520.0	316,681	8.70	8.03	18,679	429.40	40,078	1,598	57.42	....	1.99	Starboard L. P. turbine.

[a] It is assumed that all the make-up feed was evaporated.
[b] Speed of each ship was taken from its standardization speed-revolution curve.
[c] Correction was made for the difference in the running of the auxiliaries of these tests and their corresponding steam consumption tests. (See auxiliary tests.)
[d] 9.30 a. m. March 21, 1909, to 9.30 a. m. March 25, 1909.
[e] 9.45 a. m. March 29, 1909, to 11.45 a. m. March 31, 1909.
[f] 1 p. m. April 3, 1909, to 3 p. m. April 7, 1909.
[g] 10.45 a. m. April 12, 1909, to 10.45 a. m. April 13, 1909.

TABLE O.—*U. S. Scouts Birmingham, Salem, and Chester coal consumption tests*—Continued.

III-IV.—SYNOPSIS OF HORSEPOWER NECESSARY TO PROPULSION.

Number of test.	Duration of test.	Name of ship.	Speed per hour in knots.	Number of turbines in use.	Steam pressure (gage).				Vacuum (inches of mercury).		Revolutions (per minute).[a]			Horsepower.														
					Engine room.		Steam chest main engines or initial turbine.							Machinery necessary to propulsion.														Grand total (a+bb).
														Main engines or turbines S. H. P.[b]			Auxiliaries I. H. P.										
					Forward.	After.	Forward.	After.	Forward.	After.	Forward starboard.	After port.	Mean.	Forward starboard.	After port.	Both.	Augmentor.	Wet vacuum pumps.	Dry vacuum pumps.	Main air pumps.	Main circulating pumps.	Main feed pumps.	Forced-draft blowers.	Oil pumps.	Total (t+u+v+w+x+y+z+aa).		
a	b	c	d	e	f	g	h	k	l	m	n	o	p	q	r	s	t	u	v	w	x	y	z	aa	bb	cc
1	*Hrs.* 96	Birmingham	9.86	...	210.0	209.0	50.0	48.7	25.0	27.2	73.02	73.03	73.02	451.2	451.2	902.4	.....	.....	.....	11.9	17.8	19.0	12.7	.....	61.4	963.8
		Salem	9.90	2	217.9	220.4	145.3	152.0	29.6	29.1	138.23	138.23	138.23	328.6	339.8	668.4	.....	19.2	37.5	.....	24.0	25.3	8.0	1.01	115.1	783.5
		Chester	10.03	6	179.0	189.0	59.0	.....	28.8	29.0	204.04	202.36	203.20	392.0	398.4	730.4	39.4	.....	.....	19.1	17.0	29.1	.....	.55	105.1	835.1
2	50	Birmingham	15.00	...	214.0	214.0	98.0	103.0	25.6	26.8	111.88	111.95	111.91	1,495.5	1,571.7	3,067.2	.....	.....	.....	12.9	20.0	30.7	25.6	.....	89.2	3,156.4
		Salem	14.91	2	230.9	219.8	174.8	177.7	28.9	29.0	209.30	209.30	209.30	1,397.8	1,211.4	2,609.2	.....	19.5	38.2	.....	34.0	57.4	20.0	.92	170.0	2,779.2
		Chester	14.98	6	197.0	202.0	159.0	.....	28.3	28.9	313.60	310.47	284.43	1,621.3	1,580.0	3,201.3	40.6	.....	.....	14.0	32.4	40.5	.....	.61	128.1	3,329.4
3	98	Birmingham	19.83	...	209.6	206.4	173.7	172.0	23.6	26.7	149.84	149.84	149.84	3,616.2	3,504.3	7,120.5	.....	.....	.....	16.0	31.0	62.0	79.7	.....	188.7	7,309.2
		Salem	20.25	2	230.6	220.6	189.1	181.0	28.3	28.6	282.27	282.27	282.27	3,423.6	3,459.7	6,883.3	.....	22.0	37.4	.....	92.8	72.2	103.5	1.08	329.0	7,212.3
		Chester	19.90	5	172.0	176.0	147.0	.....	28.0	28.9	407.02	432.68	419.85	3,470.0	4,904.0	8,374.0	37.8	.....	.....	26.1	60.5	75.0	6.0	.82	206.2	8,580.2
4	12	Birmingham	24.00	...	250.0	238.0	229.0	226.0	25.0	26.3	189.90	189.10	189.50	7,380.0	6,556.4	13,938.4	.....	.....	.....	24.4	48.0	153.5	310.0	.....	535.9	14,474.3
	24	Salem	24.32	2	231.0	234.0	209.0	204.4	28.0	29.0	342.93	357.56	350.25	6,990.0	7,640.0	14,630.0	.....	33.2	37.2	.....	94.0	101.0	429.0	1.42	695.8	15,325.8
	24	Chester	25.08	4	212.0	216.0	204.0	201.0	28.3	29.1	560.90	555.10	558.05	10,005.0	9,627.0	19,632.0	45.8	.....	.....	29.6	128.4	156.6	101.7	1.52	463.6	20,095.6

[a] As determined from counter readings at beginning and end of test.

[b] S. H. P. of Birmingham = .94 × I. H. P.

TABLE P.—*Steam consumption of auxiliaries, in pounds, per hour.*

COMPILED FROM SUMMARY SHEETS OF COAL CONSUMPTION TESTS.

Test No.	Main air pumps.		Wet vacuum pumps.	Dry vacuum pumps.	Augmenters.	Main circulating pumps.			Steam consumption of main condenser auxiliaries (exclusive of main circulating pumps).			Steam consumption of auxiliaries necessary to propulsion.—Includes main condenser auxiliaries; forced-draft blowers; oil pumps: Salem, Chester.			Steam consumption of auxiliaries not necessary to propulsion.			Total steam consumption of auxiliaries.		
	Birmingham.	Chester.	Salem.	Salem.	Chester.	Birmingham.	Salem.	Chester.	(a) Birmingham.	(c÷d) Salem.	(b÷e) Chester.	Birmingham.	Salem.	Chester.	Birmingham.	Salem.	Chester.	(n+q) Birmingham.	(o+r) Salem.	(p+s) Chester.
	a	b	c	d	e	f	g	h	k	l	m	n	o	p	q	r	s	t	u	v
1	2,220	3,550	958	2,600	1,971	985	1,230	800	2,220	3,568	5,521	5,705	7,963	9,006	8,018	6,978	5,899	13,723	14,941	14,905
2	2,400	2,600	974	2,636	2,032	1,060	1,560	1,160	2,400	3,610	4,632	7,696	11,951	9,314	8,157	6,190	5,006	15,853	18,141	14,320
3	2,970	4,850	1,099	2,598	1,890	1,460	3,360	1,900	2,970	3,697	6,740	13,930	20,412	15,680	7,244	7,184	5,641	21,174	27,596	21,321
4	4,540	5,300	1,662	2,585	2,291	2,030	3,380	3,585	4,540	4,247	7,791	35,683	40,532	30,840	3,578	3,584	3,215	39,461	44,116	34,055

COMPARISON OF ABOVE BASED ON BIRMINGHAM (AUXILIARIES) AS 100.

1	100	160	43	132	100	100	125	81	100	160	249	100	140	158	100	87	74	100	109	109
2	100	108	41	130	100	100	147	109	100	150	193	100	155	121	100	76	61	100	114	90
3	100	163	37	137	100	100	230	130	100	124	227	100	147	113	100	99	78	100	130	101
4	100	121	37	113	100	100	165	175	100	94	172	100	113	86	100	100	90	100	112	86

* Comparison of dry vacuum pumps with augmenters as 100.

summary of the principal data will be found in tables N and O. Referring to table N, it will be seen that an attempt has been made from the weight of coal consumed, to figure boiler evaporation, as shown on these trials for comparison with results of (V) evaporative tests. The total steam used by all machinery, as recorded in column q of the table, has been carefully calculated, due allowance being made for such auxiliaries or apparatus as were intermittently used, and which required steam for their operation. It has been assumed, moreover, in this calculation (column q) that all "make-up" feed water expended (which in amount was determined from calibrated double-bottom compartments) was converted into steam, which assumption is probably not entirely correct. It will be noted, from an inspection of columns r and s (data being brought to the same basis for purposes of comparison), that a higher equivalent evaporation than that shown on boiler tests is recorded for the *Birmingham* on the 10-knot trial, and also for the *Chester* on the run at full power. No plausible explanation can be given for this, although it may be pointed out that all figures in column q are dependent upon conversion of coal, as measured, into weight, in accordance with data obtained when coaling for these trials.

Because of dissimilar propeller efficiencies, the power required to drive the different vessels (same displacement) at a given speed, differs considerably. For example, on No. 3 trial (about 20 knots) the average horsepower (shaft) of the main machinery—tabulated in table O, column s—was found to be as follows: *Birmingham*, 7,120.5; *Salem*, 6,883.3; *Chester*, 8,374. It will be obvious, therefore, that coal consumption of the machinery (either main or inclusive of all auxiliaries) per horsepower—notwithstanding the coal burned was of the same grade and quality— is not a true indication of the relative economies of the three vessels. Moreover, while coal burned per knot furnishes a more accurate means of comparison, it should be remembered that entering into a comparison of this character is the important factor of boiler efficiency, and, as well, expenditures of steam (forced-draft

blowers) necessary in conjunction with service operation of the boiler plants. Tables Q and R show boiler efficiencies under varying rates of combustion, and it will be observed that under similar conditions as to air pressure there is a wide difference in the amount of coal consumed. In this connection, it may be further pointed out, that on the 10 and 15-knot coal-consumption runs, due principally to differences in boiler design, the *Chester* steamed under natural draft, while on the other two vessels, with the number of boilers in operation, forced draft was a necessity.

Table M, above, compiled from results of coal-consumption tests, has been prepared to show available percentage of net output of steam for the three vessels. Comparison of the boiler plants should be based on this as well as boiler efficiencies.

V. Evaporative Boiler Tests.

A series of evaporative tests were made on the *Salem* and *Chester,* using the two after boilers, with a view of comparing boiler efficiencies under different rates of combustion. The boilers mentioned connect to and are the only boilers delivering gases of combustion to the after smoke pipe. The tests were carried out (in port) under as nearly as possible actual service conditions, and the boilers fired by the ships' force. It was considered unnecessary to repeat tests of this character on the *Birmingham,* because of similarity of boiler installations on this vessel and the *Salem.*

On the *Salem* four tests were made with closed fireroom under air pressures equivalent, respectively, to $\frac{1}{2}$, $1\frac{1}{4}$, $2\frac{1}{2}$ and 4 inches of water. Tests on the *Chester* covered one at natural draft with others at $\frac{1}{2}$, $1\frac{1}{4}$ and $2\frac{1}{2}$ inches of water pressure, the aim being to simulate conditions of steaming in service and to approximate to the same coal consumption per square foot of grate in the boilers of each vessel.

Before beginning each series of tests, and after the heating surfaces had been properly cleaned and examined, a test water pressure (boilers cold) was applied in order to make

sure that no leaks existed. Further cleaning of the boilers was done after each test, generally at night, as thoroughly as that was possible without hauling fires or allowing them to die out.

The main water-measuring apparatus—installed primarily to secure data on steam-consumption tests of the propelling machinery—furnished an easy and accurate method of determining the weight of water pumped to the boilers. Feed water was supplied to the forward measuring tank (fig. 1, pl. 6) and afterwards dropped to the lower or distributing tank (fig. 2, pl. 6), from which latter, through a suction connection to the feed pump, it was delivered to the boilers. All tanks prior to installation in the ships were carefully calibrated, and the weight pumped to the boilers during any test was readily calculated from its volume and temperature.

The ships' main delivery feed line was made use of to supply water to the boilers, and this was carefully examined for leaks at intervals as the tests progressed. Only the use of a comparatively short length of this line, however, was necessary, as by closing certain valves in the after fireroom the delivery of feed water forward of the boilers under test was not possible.

During tests on the *Salem,* feed water was pumped directly to the boilers from the lower or distributing tank, its temperature being somewhat raised prior to measurement, by drains blowing into the main feed tank. On the *Chester,* the feed pump discharged through the forward heater, and the water on its passage to the boilers, was raised in temperature by exhaust steam from the auxiliaries in use.

Steam generated by the boilers under test was supplied to the main steam line, and so much utilized as was necessary to operate port auxiliaries; any excess was disposed of either through specially rigged piping directed overboard (*Salem*) or delivered to the main condenser (*Chester*) through bleeder-pipe connections.

Moisture in the steam was determined by Barrus throttling calorimeters, one of these instruments being fitted to each

boiler and connected to main stop-valve chambers on boiler side of valves. The distance of calorimeters from boiler drums was about 12 inches.

Coal was weighed on platform scales in the firerooms, and at the beginning and end of each test, fires were carefully leveled so as to have, as nearly as could be judged, the same amount on the grates. It may be stated here that in planning these tests it was the intention to use the same kind and grade of coal in all, but on account of the movement of the vessels this proved inconvenient. The two series were made, therefore, with Georges Creek (*Salem*) and Eureka (*Chester*) coal, the vessels being temporarily at the Boston and New York Navy Yards, respectively, when the tests were carried out.

Considerable ash and cinder were blown out of the smoke pipes in all tests except No. 4 (natural draft) on the *Chester*. Because of this, determinations of weight of ashes hauled from the ash pans were considered valueless, and these weights, in consequence, have been figured from results of chemical analyses of the coal used.

Temperature of smoke-pipe gases were taken (about 35 feet above grates) by mercurial and electric pyrometers. In tabulating results, however, readings only of the mercurial pyrometer are recorded, except in test No. 3 on the *Chester*. In this test, flame was observable, at frequent intervals, through the aperture into which the pyrometers fitted, a circumstance which did not occur during other tests, and the electric pyrometer only was used on account of high temperature.

Analyses of flue gases on each test were made by Orsatt apparatus, at hourly periods on the *Salem* and at half-hourly intervals on the *Chester*. The average of all such analyses for each test is recorded in the tables.

A summation of the data of all tests is given in tables Q and R, and it will be noted, from a comparison of results, that the boilers installed on the *Salem*, under approximately similar conditions as to consumption of coal, show a higher thermal

TABLE Q.—*U. S. S. Salem.*

Number of test.	1.	2.	3.	4.
1. Date of test	July 17, 1909	July 18, 1909	July 19, 1909	July 20, 1909
2. Duration of test.....hours	12	10	8	6
3. Kind of fuel	Georges Creek.	Georges Creek.	Georges Creek.	Georges Creek.
4. Kind of start	Continuous fires.	Continuous fires.	Continuous fires.	Continuous fires.
5. State of weather	Fair.	Fair.	Fair.	Fair.
AVERAGE PRESSURES.				
6. Barometer.....inches	29.90	29.88	29.76	30.24
7. Steam pressure by gage.....pounds	242	221	245	247
8. Force of draft at base of pipe.....inches of water	−0.4	−0.6	−0.7	−0.7
9. Force of draft in furnace.....do	0.16 0.17	0.81 0.78	1.59 1.47	2.67 2.66
10. Force of draft in fireroom.....do	0.5 0.5	1.25 1.25	2.5 2.5	4.0 4.0
11. Revolutions of blower	195	294	405	512
AVERAGE TEMPERATURES.				
12. External air.....°F.	77	79	72	66
13. Fireroom.....do	115	110	101	95
14. Steam.....do	403	396	405	405
15. Feed water entering heater.....do	Not in use.	Not in use.	Not in use.	Not in use.
16. Feed water entering economizer.....do	None.	None.	None.	None.
17. Feed water entering boiler.....do	127	90	117	111
18. Air entering ash pit.....do	115	110	101	95
19. Escaping gases from boiler.....do	476	558	632	684
20. Escaping gases from economizer.....do	None.	None.	None.	None.
FUEL.				
21. Kind of	Georges Creek.	Georges Creek.	Georges Creek.	Georges Creek.
22. Weight of wood used in lighting fires.....pounds				
23. Weight of coal as fired.....do	23,800	35,000	40,650	42,600
24. Moisture in coal.....per cent	1.44	Ave. 1.11	0.78	Ave. 1.11
25. Weight of dry coal consumed.....do	23,458	34,612	40,333	42,127
26. Weight of ash by analysis.....pounds	1,560	2,721	3,601	3,016
27. Weight of combustible consumed.....do	21,898	31,891	37,332	39,111
28. Per cent of ash by analysis in dry coal	6.65	7.86	7.44	7.16
FUEL PER HOUR.				
29. Coal consumed per hour.....pounds	1,983	3,500	5,081	7,100
30. Dry coal consumed per hour.....do	1,955	3,461	5,042	7,021
31. Combustible consumed per hour.....do	1,825	3,189	4,667	6,519
32. Coal consumed per hour per square foot G. S.....do	17.09	30.17	43.80	61.20
33. Dry coal consumed per hour per square foot G. S.....pounds	16.85	29.84	43.47	60.53
34. Combustible consumed per hour per square foot G. S.....pounds	15.73	27.49	40.23	56.20
35. Coal per hour per square foot H. S.....do	0.313	0.552	0.802	1.121
36. Dry coal per hour per square foot H. S.....do	0.308	0.546	0.796	1.109
37. Combustible per hour per square foot H. S....do	0.288	0.503	0.737	1.029

TABLE Q.—*U. S. S. Salem*—Continued.

Number of test.	1.	2.	3.	4.
QUALITY OF STEAM.				
38. Per cent of moisture in steam	0.241	0.646	0.44	0.671
39. Degrees of superheating	None.	None.	None.	None.
40. Quality of steam (dry steam=100)	99.759	99.354	99.56	99.329
WATER.				
41. Total weight of water fed to boiler [a] ..... pounds	230,663	298,836	347,349	329,863
42. Water actually evaporated, corrected for quality of steam (40 by 41) ..... pounds	230,127	296,905	345,821	327,666
43. Factor of evaporation	1.15	1.186	1.161	1.167
44. Equivalent water evaporated into dry steam from and at 212° (42 by 43) ..... pounds	264,646	352,129	401,152	382,386
WATER PER HOUR.				
45. Water evaporated per hour, corrected for quality of steam ..... pounds	19,177	29,690	43,228	54,611
46. Equivalent evaporation from and at 212°	22,054	35,213	50,144	63,731
47. Equivalent evaporation from and at 212° per square foot G. S. ..... pounds	190	303	432	549
48. Same per square foot of heating surface ..... do	3.48	5.56	7.92	10.06
ECONOMIC RESULTS.				
49. Water apparently evaporated under actual conditions per pound of coal as fired (41÷23) ..... pounds	9.69	8.54	8.54	7.71
50. Apparent equivalent evaporation from and at 212° per pound of coal (including moisture) 44÷23, pounds	11.12	10.06	9.84	8.94
51. Equivalent evaporation from and at 212° per pound of dry coal (44÷25) ..... pounds	11.28	10.17	9.95	9.07
52. Equivalent evaporation from and at 212° per pound of combustible (44÷27) ..... pounds	12.08	11.04	10.74	9.77
EFFICIENCY.				
53. Efficiency of boiler; heat absorbed by the boiler per pound of combustible divided by the heat value of one pound of combustible ..... per cent	75.49	69.51	66.54	60.96
54. Efficiency of boiler, including grate; heat absorbed by the boiler per pound of dry coal, divided by the heat value of one pound of dry coal				
REMARKS AND OBSERVATIONS.				
55. Principal data taken every	Hour.	Hour.	Hour.	Hour.
56. Percentage of smoke as observed		No estimate made.		
57. Average thickness of fires	4" to 5"	5" to 6"	6" to 7"	8" to 9
58. Efficiency of firemen; expert, average, or poor		Good average fireman.		

[a] Corrected for inequality of water level and steam pressure at beginning and end of test.

TABLE Q—*U. S. S. Salem*—Continued.

FUEL AND GAS ANALYSES.

PROXIMATE ANALYSIS OF FUEL.

Number of test.	1.	2.	3.	4.
	Per cent.	*Per cent.*	*Per cent.*	*Per cent.*
Fixed carbon	73.76	71.82	72.68	72.85
Volatile matter	18.93	18.67	18.61	18.90
Moisture	0.82	1.38	1.19	1.05
Ash	6.49	8.13	7.52	7.20
Total	100.00	100.00	100.00	100.00
Sulphur separately determined	0.927	1.04	0.95	0.93

ULTIMATE ANALYSIS OF DRY FUEL.

Number of test.	1.	2.	3.	4.
	Per cent.	*Per cent.*	*Per cent.*	*Per cent.*
Carbon (C)	83.10	81.40	81.69	81.85
Hydrogen (H)	4.57	4.62	4.63	4.54
Oxygen (O)	3.38	3.85	4.07	4.22
Nitrogen (N)	1.37	1.22	1.22	1.29
Sulphur (S)	0.93	1.05	0.95	0.94
Ash	6.65	7.86	7.44	7.16
Total	100.00	100.00	100.00	100.00
Moisture in sample of fuel as received	0.82	1.38	1.19	1.05

CALORIFIC VALUE OF FUEL.

Number of test.	1.	2.	3.	4.
	B. T. U.	*B. T. U.*	*B. T. U.*	*B. T. U.*
Calorific value by calorimeter, per pound of dry coal	14,430	14,135	14,419	14,370
Calorific value by calorimeter, per pound of combustible	15,455	15,339	15,580	15,477
Calorific value by analysis, per pound of dry coal	14,705	14,451	14,482	14,436
Calorific value by analysis, per pound of combustible	15,751	15,682	15,645	15,548

ANALYSES OF DRY GASES.

Number of test.	1.	2.	3.	
	Per cent.	*Per cent.*	*Per cent.*	*Per cent.*
Carbon dioxide (CO_2)	7.16	8.30	9.83	10.10
Oxygen (O)	9.06	11.03	10.00	9.15
Carbon monoxide (CO)	0.88	0.28	1.24	1.23
Nitrogen (N) (by difference)	82.90	80.39	78.33	79.52
Total	100.00	100.00	100.00	100.00

TABLE R.—*U. S. S. Chester.*

Number of test.	1.	2.	3.	4.
1. Date of test	Feb. 2, 1909.	Feb. 3, 1909.	Feb. 4, 1909.	Feb. 5, 1909.
2. Duration of test, hours	10	8	6	12
3. Kind of fuel	Eureka.	Eureka.	Eureka.	Eureka.
4. Kind of start	Continuous fires.	Continuous fires.	Continuous fires.	Continuous fires.
5. State of weather	Fair.	Fair.	Fair.	Fair.
AVERAGE PRESSURES.				
6. Barometer, inches				
7. Steam pressure by gage, pounds	258.5	257	258	252
8. Force of draft at base of pipe, inches of water	−0.4	−0.6	−0.7	−0.4
9. Force of draft in furnace, do	0.0 0.02	0.42 0.45	1.1 1.22	−0.2 −0.2
[1]10. Force of draft in fireroom, do	0.5 0.5	1.25 1.25	2.5 2.5	Natural draft.
11. Revolutions of blower	Not taken.			
AVERAGE TEMPERATURES.				
12. External air, °F.	Do.			
13. Fireroom, do	77	65	62	88.7
14. Steam, do	408.5	408	408.5	406.5
15. Feed water entering heater, do	103	87	86	103
16. Feed water entering economizer, do	None.	None.	None.	None.
17. Feed water entering boiler, do	143	115	98	215
18. Air entering ash pit, do	77	65	62	88.7
19. Escaping gases from boiler, do	655	722	973	602
20. Escaping gases from economizer, do	None.	None.	None	None.
FUEL.				
21. Kind of	Eureka.	Eureka.	Eureka.	Eureka.
22. Weight of wood used in lighting fires, pounds	None.	None.	None.	None.
23. Weight of coal as fired, do	33,200	36,450	38,550	25,875
24. Moisture in coal, per cent	1.5	2.75	2.0	Ave. 1.92
25. Weight of dry coal consumed, pounds	32,702	35,448	37,779	25,379
26. Weight of ash by analysis, do	2,093	2,091	2,875	1,718
27. Weight of combustible consumed, do	30,609	33,357	34,904	23,663
28. Per cent of ash by analysis in dry coal	6.40	5.90	7.61	6.76
FUEL PER HOUR.				
29. Coal consumed per hour, pounds	3,320	4,556	6,425	2,156
30. Dry coal consumed per hour, do	3,270	4,430	6,296	2,115
31. Combustible consumed per hour, do	3,061	4,170	5,817	1,972
32. Coal consumed per hour per square foot G. S., do	28.62	39.28	55.39	18.58
33. Dry coal consumed per hour per square foot G. S., pounds	28.19	38.19	54.28	18.23
34. Combustible consumed per hour per square foot G. S., pounds	26.38	35.95	50.15	17.00
35. Coal per hour per square foot H. S., do	0.622	0.853	1.203	0.403
36. Dry coal per hour per square foot H. S., do	0.612	0.829	1.179	0.396
37. Combustible per hour per square foot H. S., do	0.573	0.781	1.089	0.369
QUALITY OF STEAM.				
38. Per cent of moisture in steam	0.173	0.31	0.401	0.143
39. Degrees of superheating	None.	None.	None.	None.
40. Quality of steam (dry steam=100)	99.827	99.69	99.599	99.862

TABLE R.—*U. S. S. Chester*—Continued

Number of test.	1.	2.	3.	4
WATER.				
41. Total weight of water fed to boiler a.....pounds..	273,896	287,031	265,766	237,759
42. Water actually evaporated, corrected for quality of steam (40 by 41).....pounds..	273,420	286,141	264,700	237,430
43. Factor of evaporation.....	1.134	1.163	1.180	1.058
44. Equivalent water evaporated into dry steam from and at 212° (42 by 43).....pounds..	309,605	332,381	312,160	250,963
WATER PER HOUR.				
45. Water evaporated per hour, corrected for quality of steam.....pounds..	27,342	35,769	44,117	19,786
46. Equivalent evaporation from and at 212°.....	30,960	41,548	52,026	20,914
47. Equivalent evaporation from and at 212° per square foot G. S.....pounds..	267	358	448	180
48. Same per square foot of heating surface.....do....	5.80	7.78	9.74	3.91
ECONOMIC RESULTS.				
49. Water apparently evaporated under actual conditions per pound of coal as fired (41÷23)..pounds..	8.25	7.88	6.89	9.19
50. Apparent equivalent evaporation from and at 212° per pound of coal (including moisture) 44÷23 .....pounds..	9.33	9.12	8.10	9.70
51. Equivalent evaporation from and at 212° per pound of dry coal (44÷25).....pounds..	9.46	9.38	8.26	9.88
52. Equivalent evaporation from and at 212° per pound of combustible (44÷27).....pounds..	10.11	9.96	8.94	10.61
EFFICIENCY.				
53. Efficiency of boiler; heat absorbed by the boiler per pound of combustible divided by the heat value of 1 pound of combustible.....per cent..	62.48	61.59	55.71	65.68
54. Efficiency of boiler, including grate; heat absorbed by the boiler per pound of dry coal, divided by the heat value of 1 pound of dry coal.	No analysis was made of the ashes nor of the cinder blown out of the stack.			
REMARKS AND OBSERVATIONS.				
55. Principal data taken every.....	½ hour.	½ hour.	½ hour.	½ hour.
56. Percentage of smoke as observed.....	No estimate made.			
57. Average thickness of fires.....	5" to 6"	6" to 7"	8" to 9"	4" to 5"
58. Efficiency of firemen; expert, average, or poor.....	Good average firemen.			

a Corrected for inequality of water level and steam pressure at beginning and end of test.

TABLE R.—*U. S. S. Chester*—Continued.

FUEL AND GAS ANALYSES.

PROXIMATE ANALYSIS OF FUEL.

Number of test.	1.	2.	3.	4.
	Per cent.	*Per cent.*	*Per cent.*	*Per cent.*
Fixed carbon	76.45	76.84	75.81	76.61
Volatile matter	16.17	15.75	15.51	15.36
Moisture	.52	.97	.78	.89
Ash	6.87	6.44	8.20	7.14
Total	100.00	100.00	100.00	100.00
Sulphur separately determined	1.36	1.53	1.70	1.14

ULTIMATE ANALYSIS OF DRY FUEL.

Number of test.	1.	2.	3.	4
	Per cent.	*Per cent.*	*Per cent.*	*Per cent.*
Moisture in sample of fuel as received	0.52	0.97	0.78	0.89
Carbon (C)	83.41	84.06	81.82	83.27
Hydrogen (H)	4.42	4.39	4.31	4.33
Oxygen (O)	3.30	3.03	3.49	3.44
Nitrogen (N)	1.10	1.08	1.06	1.05
Sulphur (S)	1.37	1.54	1.71	1.15
Ash	6.40	5.90	7.61	6.76
Total	100.00	100.00	100.00	100.00
Moisture in sample of fuel as received	0.52	0.97	0.78	0.89

CALORIFIC VALUE OF FUEL.

Number of test.	1.	2.	3.	4.
	B. T. U.	*B. T. U.*	*B. T. U.*	*B. T. U.*
Calorific value by calorimeter, per pound of dry coal	14,629	14,697	14,330	14,548
Calorific value by calorimeter, per pound of combustible	15,627	15,647	15,498	15,600
Calorific value by analysis, per pound of dry coal	14,673	14,744	14,370	14,576
Calorific value by analysis, per pound of combustible	15,674	15,668	15,561	15,630

ANALYSES OF DRY GASES.

Number of test.	1.	2.	3.	4.
	Per cent.	*Per cent.*	*Per cent.*	*Per cent.*
Carbon dioxide (CO_2)	6.33	7.16	9.65	9.14
Oxygen (O)	15.04	11.60	9.43	9.81
Carbon monoxide (CO)	0.34	0.76	0.25	0.53
Nitrogen (N) (by difference)	78.29	80.48	80.69	80.52
Total	100.00	100.00	100.00	100.00

efficiency than the type on the *Chester*. This, unquestionably, is attributable, in part, to the larger percentage of tube-heating surface in the *Salem's* boilers, but more particularly to the tortuous passage traversed by the gases of combustion.

Graphical representations, based on results of evaporative tests, and coal-consumption tests, as well are shown on plate 14.

VI. Tests of Auxiliary Machinery.

(Tables 2–32; Summary Table T)

Immediately after installation of auxiliary measuring tanks on each ship, a series of tests were run to establish, under varying conditions, the steam consumption of individual auxiliaries. These tests, carried out at the Boston Navy Yard, were the first to be undertaken, and results in detail are set forth in tables 2 to 32.

The method adopted was to isolate and condense the exhaust steam from the auxiliary under test, using the forward engine-room auxiliary condenser, and to ascertain the weight of the resulting water of condensation. To facilitate securing the data sought, temporary atmospheric exhausts were rigged to such auxiliaries as were likely to be run either constantly or intermittently during the tests, and which ordinarily exhausted into the auxiliary exhaust line. These included an auxiliary-boiler feed pump, steam cylinder operating air and circulating pumps of auxiliary condenser, ice machines, etc. Dynamo installations, as previously pointed out, were provided with independent condensing plants, so that no special provision to handle exhaust steam from these was necessary.

In the three ships the auxiliary exhaust line, in a fore-and-aft direction, runs almost parallel to the center line, with suitable branch connections to each of the various auxiliaries. This line of piping was used to convey exhaust steam from the auxiliary being tested to the engine-room auxiliary condenser, from which, in the form of water, it was delivered by the air pump to the auxiliary measuring tanks. Plate 6, figure 3, shows the construction of these auxiliary tanks, and

it will be seen then, for each ship, but one tank, of rectangular section, provided with a central watertight diaphragm, was required. The division of each into two separate compartments constituted, in reality, two measuring tanks of about equal capacity, into either of which the water of condensation could be directed, as desired, by a swinging gooseneck.

No special provision, other than fitting certain appliances (gauges, counters and indicator gears), was made to prepare any auxiliary for test, a superficial examination being alone considered necessary when the auxiliary appeared to be in satisfactory working order. It was the aim to operate each auxiliary under conditions met with in service, but examination or special fitting of steam-cylinder packing to influence steam economy was not attempted. Counters were generally fitted in all cases where not permanently installed, to determine accurately the revolutions, or double strokes, during the test; in addition, steam gauges were installed on steam chests, and in some tests, such, for example, as those of feed, flushing and ash-ejector pumps, gauges were also fitted to indicate discharge pressures. Moreover, to assimilate actual service conditions, a back pressure of about 6 pounds above the atmosphere was maintained in the auxiliary exhaust line, during all tests, by manipulating a regulating valve governing inlet to auxiliary condenser, the pressure noted being that usually carried when exhaust steam from the auxiliaries is utilized in the feed-water heaters, as is the custom in cruising.

It was usual, before the commencement of an auxiliary test, to subject the system as a whole to a preliminary test for tightness. This merely involved inspection of the air-pump discharge, after starting auxiliary condenser, the absence of water issuing therefrom being accepted as indicating not only that the auxiliary exhaust line was clear, but that the condenser was free from salt-water leaks as well. The auxiliary to be tested was next started, but before beginning to record data, sufficient time—usually one-half hour or more—was allowed to elapse to insure normal working conditions. During this period the water of condensation was

permitted to run directly through one of the two compartments of the measuring tank to the feed tank below. At the instant of beginning the test the swinging goose neck, attached to the air-pump discharge, was shifted and directed into the other empty compartment of the measuring tank, the gate valve at the bottom of this having been previously closed.

The water of condensation was thus collected, and by aid of a graduated scale and water-gauge glass, attached to each compartment, the height in inches, as the tests progressed, as well as full tanks, could be noted and recorded. Opportunity offered, as each compartment was being filled, to prepare the remaining one for use, so that tests could be carried on for any length of time, and without interruption, by alternately shifting goose-neck connection of air-pump discharge from one to the other as the compartments filled. In some tests, as will be observed from the tables, it was not considered necessary to collect an entire tank of condensed water, while others were continued to include several tanks. Boiler and steam-chest pressures and pressure in auxiliary exhaust line, with the data necessary to compute power, are recorded in appropriate columns in the various tables.

It will be apparent from a survey of the tabulated data that essentially the same mode of procedure was adopted in all auxiliary tests. At the start of each the water of condensation was directed into one compartment of the measuring tank, and as the test advanced, the elapsed time to fill the various heights (usually by progressive intervals of 5 or 10 inches) up to "tank full," together with the temperature of water at each reading, was recorded. From tank-calibration tables, making due allowance for temperature—which correction generally was immaterial—the weight of collected water up to any height was readily ascertained. Regularity in time interval to fill equal increments in height (each tank compartment being closely of the same rectangular section up to about 45 inches) usually indicated regularity in steam consumption of the auxiliary under test. This, how-

ever, was not necessarily the case, as it was noticed, particularly when the measuring tank filled slowly and therefore the quantity of steam being condensed was comparatively small, that gulps of water were frequently delivered at irregular intervals. The cause of this may be traced to pockets in the exhaust line of piping or possibly to failure of the air pump to deliver small quantities of water with unvaried regularity.

To assist in analyzing auxiliary tests, a number of curves were plotted from the data. Curves constructed using steam consumption per hour as abcissae, with ordinates representing revolutions (or double strokes) per minute, multiplied by the weight of a cubic foot of steam at the average steam-chest pressure, proved of substantial value in differentiating the steam used by individual auxiliaries on main steam-consumption trials, where the condensed exhaust steam from all auxiliaries, as a whole, was measured; also, in coal-consumption trials, to calculate approximately the steam used by the various auxiliaries in operation. It should be observed that points of curves with co-ordinates as above noted, seemed to lie generally in a straight line, and the conclusion was reached that this invariably follows for those auxiliaries with constant cut-off and which remain in a constant working condition.

As stated, during main steam-consumption trials (II), the amount of condensed exhaust steam as a whole from the auxiliaries in operation, was ascertained by measurement in the auxiliary tanks. On these tests and other sea trials as well, sufficient data were recorded also to enable a close approximation of power and steam consumption to be made of each auxiliary by comparison with results of auxiliary tests; synopsis tables of sea trials, heretofore given, contain estimates for auxiliaries obtained in this way.

In comparing trial performances of the three vessels, it has been necessary to convert indicated into shaft horsepower for the main propelling engines of the *Birmingham,* but a similar conversion, as applied to the auxiliaries, has not been

attempted in any case. While for the total developed power of each ship this involves adding shaft horsepower of the propelling machinery and indicated power of the auxiliaries, which is undesirable from the standpoint of unlike units, nevertheless this method seems more satisfactory than entering into an intricate calculation as to the relation between the two units for each auxiliary, which, however carefully done, could only be regarded finally as an approximation.

Table S gives principal dimensions of the various auxiliaries, and it will be noticed that in the machinery equipment of each vessel there are several of the same size and make. Tests of economy, as a rule, were undertaken only on one of each size, but the results obtained are considered equally applicable to all of the same size when similarly used.

The auxiliaries installed on each vessel, with respect to location, may be classed under the following three heads: 1, engine-room auxiliaries; 2, fireroom auxiliaries; and 3, other or outside auxiliaries. This classification, which is observed in listing the auxiliaries in table S, is followed also in the notes below, which refer to tests of the different auxiliaries.

NOTES ON AUXILIARY TESTS.

I. ENGINE-ROOM AUXILIARIES.

Main Air Pumps.

(Tables 2 and 3.)

Table 2 shows the data collected on four tests of the after main air pump of the *Birmingham*. This pump is of the Blake featherweight variety, double-acting (both steam and water), simplex type, with two steam and two water cylinders. Each steam piston gives motion to its own water piston, and the arrangement is such that one half of the pump may be in operation while the remaining half is at rest. When cruising at low powers, and to effect steam economy, it is customary to use but one side of the pump. In tests 1, 2 and 3 one steam cylinder only was in use; in test 4 both steam cylinders were in operation at the same speed. During all tests the line of piping, which collects the discharge from all traps, was open to main condenser; the air pump handled no other water; main injection and outboard delivery valves were wide open, but circulating pump was not in operation. Pump made full (18 inches) stroke on all tests. In estimating and apportioning to each auxiliary the weight of steam used on the various sea trials a consumption of 186 pounds per indicated horsepower per hour has been taken for this pump.

The *Chester's* main air pumps are of the Blake, beam, vertical type, with two steam (double-acting) cylinders, having a common steam chest and two (single-acting) water cylinders. Main trap-discharge line was open to main condenser during all tests, as well as main injection and discharge valves; circulating pump was not in use. Stroke of pump during tests was 18½ inches. The same steam consumption as stated for the *Birmingham's* air pump, per unit of power per hour, has been taken for this pump in calculations relating to individual steam expenditures of auxiliaries on sea trials.

Augmenters.

For the purpose of further adding to the vacuum produced by the air pumps, an augmenter is installed on the *Chester* in connection with each main condenser. When the ship is under way these are in constant use, the steam thus expended being practically the same in amount for all speeds. Tests in port, with the propelling machinery stopped and extending over a period of two and one-half hours, were carried out to determine the weight of steam used, the water of condensation being delivered to and measured in the main feed tanks. With an average steam pressure of 184 pounds (gauge) and steam-supply valve open to the augmenters the usual number of turns, the weight per hour was found to be: Forward, 904; and, aft, 992 pounds. On (II) steam-consumption trials it will be apparent that all steam used by the augmenters necessarily forms part of the measured water of condensation from the main propelling machinery. An equivalent indicated horsepower, based upon a steam consumption of 50 pounds per horsepower per hour, has been assigned to each augmenter in estimating power of auxiliaries on sea trials.

Wet (Centrifugal) and Dry (Rotary) Vacuum Pumps.
(Tables 4 and 5.)

Because of the importance, with turbine machinery, of maintaining a high vacuum at all times, a wet and dry-vacuum pump form part of each main condenser equipment of the *Salem*. The first mentioned is of the centrifugal vertical type, driven by a Curtis 2-stage turbine, which is provided with three steam nozzles, each of tapering rectangular (0.0375 square inch area at the smallest part) cross section. Two of the three steam nozzle openings are fitted with control valves, the remaining one being constantly open. The pump speed is regulated by a governor, and steam, after passing the throttle, is admitted to the turbine through one, two, or three nozzles, as found requisite to handle efficiently the water of condensation resulting from entry of exhaust steam into the condenser. No power test on this type of pump was attempted. Tests to fix steam economy with one and two nozzle openings—discharge from various traps only entering the main condensers—are recorded in Table 4. It may be remarked here that on all steam-consumption tests of the *Salem's* propelling machinery, the steam necessary to operate these pumps could not well be separated from that used by the main turbines; for although proper connections to the auxiliary exhaust line

were installed, unsatisfactory working resulted with vessel under way, when so operated against a back pressure above the atmosphere. The exhaust from each pump, therefore, was directed into its main condenser. In calculations relating to power of auxiliaries on sea trials, an equivalent indicated horsepower has been assigned to each wet-vacuum pump based upon a steam consumption of 50 pounds per hour per I.H.P.

For the removal of air, etc., each main condenser is provided with a rotary dry-vacuum pump, consisting of one air cylinder, operated by a single-cylinder steam engine. Results of tests of this auxiliary are tabulated in Table 5.

Main Circulating-Pump Engines.

(Tables 6, 7 and 8.)

A compound engine operates a centrifugal pump to supply cooling water to each main condenser. These installations on the *Birmingham* and *Salem* are alike in all particulars, except that the engine shaft of the latter carries a bilge pump worked by a geared crank. H.P. cylinder diameter and stroke of the *Chester's* engine are somewhat larger than similar dimensions of the engine fitted in the other two ships, permitting in consequence the development of greater power. In all tests the main injection and outboard delivery valves were wide open.

Oil-Circulating Pumps.

To insure proper lubrication of main turbine bearings and thrust blocks, oil-circulating pumps, operating on a closed system, are installed. The *Salem* is provided with two such pumps in each engine room, but under ordinary cruising conditions the operation of one pump in connection with each turbine is sufficient. Oil is continuously supplied from closed tanks installed in the upper engine-room hatches, to the various bearings, under a gravity head of about 20 feet, and drained therefrom to coolers located near the pumps. The purpose of the coolers is to reduce the temperature of oil flowing to them, each being arranged by suitable connections to the main circulating pumps for circulation of cooling water. Each pump has a suction to the coolers with a discharge to the supply tank.

The *Chester* has but one oil-circulating pump in each engine room, either or both of which may be used as desired on the oiling system. Oil is supplied to the bearings under pressure (about 10 pounds), and the arrangement of coolers, etc., in essential particulars is similar to that described on the *Salem*.

No steam-consumption tests were carried out on these pumps. A test, however, was begun on the *Chester*, and discontinued after it became apparent that the amount of steam used under ordinary conditions of working was extremely small, and could only be accurately determined by continuing the test over an abnormal time interval. Steam used by these pumps on main steam and coal consumption trials, as as well as power, has been estimated from results of other pump tests; a figure of 200 pounds per I.H.P. per hour has been assumed in all cases.

Main Feed Pumps.

(Tables 9, 10 and 11)

In the original design, two main feed pumps for each ship, of the size shown in Table S, were estimated to be a sufficient capacity to supply the boiler installations with the requisite quantity of feed water when developing maximum power. Upon official trials, however, a maximum speed and power greater than that planned, especially with the turbine-propelled ships, was developed, making it desirable, at least viewed from the standpoint of safety, to provide an additional feed pump. This led eventually to fitting a third feed pump on each vessel of the same size as the other two, although these pumps were not finally installed until just prior to the coal-consumption runs. In the meantime, and before the main steam-consumption tests were undertaken, each vessel was provided with a temporary additional feed pump for emergency use.

Tables 9, 10 and 11 contain results of the various tests carried out, the pumps in all cases making full stroke. It should be observed that during all tests, in order to assimilate service conditions, the pumps delivered water against a pressure approximating that generally carried in the main feed line when steaming at sea. This was accomplished by arranging the pumps to draw from double bottom compartments and redelivering the fresh water to the same place, a valve on the delivery side of pumps being regulated to maintain the desired pressure.

Fire and Bilge Pumps.

(Tables 12, 13 and 14.)

These tests were conducted with fire and bilge pumps drawing water from and returing it to the sea, under varying discharge pressures, as shown in the tables. In all tests, pumps made full stroke.

Engine-Room Auxiliary Condenser.

(Tables 15 and 16)

One cylindrical horizontal auxiliary condenser is located in the forward engine room of each vessel. Attached pumps (air and circulating) are horizontal and operated by a single steam cylinder.

II. FIREROOM AUXILIARIES.

Auxiliary Feed Pumps.

(Tables 17 and 18.)

In tests of these pumps fresh water was pumped from and returned to the double-bottom compartments, the discharge pressure being regulated in the same manner and for the same reason as heretofore explained under "Main feed pumps." Pumps made full stroke in all tests.

Fire and Bilge Pumps (Used in Ejecting Ashes).

(Tables 19 and 20.)

In each fireroom there is installed a fire and bilge pump, which may be used, in conjunction with a hopper arrangement, for ejecting ashes. At such times, to insure successful working, it is essential to

carry a comparatively high pump discharge pressure. Tests recorded were made under normal conditions of service with the pumps subjected to this use, and were so regulated as to make designed stroke on all tests.

Forced-Draft Blower Engines.

(Tables 21, 22 and 23.)

On the *Birmingham* and *Salem* the forced-draft blower installations consist of six separate directly-connected blowers, two to each fireroom compartment, driven by independent, 2-cylinder simple engines. Table 21 contains data of five tests (one blower and engine) on the *Birmingham*, operating under different steam-chest pressures, free escape of air supplied the fireroom being permitted through discharge opening of the blower at rest. In these tests care was taken to maintain at all times precisely the steam-chest pressure tabulated.

Table 22 contains the results of the six tests on the *Salem* with two blowers in use, under air pressures varying from one-half to four inches of water, the fireroom being closed airtight, and escape of air governed by opening boiler furnace doors. It was the aim to carry a constant air pressure in each of these tests, the blower engines being speeded or slowed, as required, to accomplish this end.

Eight tests on the *Chester's* forced-draft blower engine, recorded in Table 23, were carried out in the same way as noted for the *Salem*.

III. OUTSIDE AUXILIARIES.

Distiller Circulating Pumps.

(Tables 24 and 25.)

Two distiller circulating pumps, either or both of which may be used, in addition, for flushing purposes, are installed in each ship. Tests were with the pumps delivering sea water to the flushing main only, the evaporating plant being shut down. With the latter in operation it is the ordinary practice to use but one pump to supply cooling water to the distillers, while the other operates on the flushing line.

Evaporating Plant.

(Table 26.)

The fresh-water distilling plant on each vessel consists of four evaporators (single effect) and four distillers, having a combined capacity of about 16,000 gallons potable water per day. Four pumps are fitted in connection with each plant, as follows: Two distiller circulating, one evaporator feed, and one fresh water.

Tests of four hours' duration were carried out on each ship, with two evaporators and accessories in use, the object being to ascertain the weight of fresh water distilled per pound of boiler steam used, with the plant operating under normal service conditions. The condensed steam collected included that discharged from evaporator coils, as well as the exhaust from various (one distiller circulating, evaporator feed and fresh water) pumps.

Electric Plants.

(Tables 27, 28 and 29.)

Each vessel is provided with three 32-kilowatt General Electric generating sets, compound wound, of the multipolar type, and each driven by a direct-connected compound engine. In addition to lighting the vessel, current is supplied for electrical operation of a number of auxiliaries, such, for example, as deck winches, hull-ventilation fans, workshop machinery tools, etc. An auxiliary condenser, similar in type to that fitted in the forward engine room, but smaller in dimensions, is installed for the exclusive use of each dynamo plant.

Tables 27, 28 and 29 contain data covering tests of one engine on each vessel under various electric loads. Indicator cards were secured on almost all tests, but the results from those taken on the *Birmingham* and *Salem* exhibited such obvious inconsistencies (due probably to inaccurate indicator springs or faulty indicator connections) as to justify their rejection. Cards from the *Chester's* engine apparently give fairly consistent and reliable results.

Tests to establish the quantity of steam necessary for operation of the air and circulating pumps of dynamo auxiliary condensers were not made, as the steam cylinder driving these pumps is of the same diameter and stroke as fitted to auxiliary condensers in engine rooms.

Refrigerating Plants.

(Tables 30 and 31.)

Two vertical, 1-ton Allen dense-air (each having one steam, one compressor and one expander cylinder) ice machines are installed on each ship. As proper gear was lacking for the motion, no indicator cards were secured from the steam cylinder during the tests.

Steering Engines.

(Table 32.)

A Williamson steam-steering engine (two-cylinder, simple, horizontal) is installed on the *Birmingham* and *Salem*.. The *Chester* is provided with a Hyde steering engine of similar type and principal dimensions, the power, in all ships being transmitted to the rudder head by a right-and-left-hand screw shaft.

Table 32 gives the weight of steam wasted per hour by the steering engine of each ship (due to leakage) with throttles open (change valves in midposition) but no movement of the engines. Results are also tabulated (main propelling machinery not in use) when helm is slowly shifted, at precisely two-minute intervals, 3 degrees to starboard, then the same angle to port, and immediately afterwards brought to rest amidships.

TABLE S.—*Principal dimensions of auxiliaries.*

Identification letter.	Auxiliaries.		Birmingham.					Salem.					Chester.				
	Name.	Type.	No.	Diameter of cylinders.				No.	Diameter of cylinders.				No.	Diameter of cylinders.			
				No.	Steam.	Water.	Stroke.		No.	Steam.	Water.	Stroke.		No.	Steam.	Water.	Stroke.
a	Main air pumps	Double acting	2	2	12″	22″ air.	18″										
		Single acting	2										2	2	14″	35″ air.	21″
b	Augmenter												2	¾-inch nozzle.			
c	Wet vacuum pump	Two stage vertical centrifugal.						2	Two stage 27″ rotor turbine pump, rotor 12″ d.								
d	Rotary dry vacuum	Vertical with fly wheel						2	1	10″	24″ air.	18″					
e	Main circulating	Centrifugal	2	2	8″ / 16″	36″ runner.	8″	2	2	8″ / 16″	36″ runner.	8″	2	2	10″ / 16″	42″ runner.	10″
f	Main feed pumps	Vertical simplex	3	1	15″	10″	15″	3	1	15″	10″	15″	3	1	15″	10″	15″
g	Reserve feed pumps	do	1	1	6″ Blake.	4″ duplex.	7″	1	1	3½″	4″	4″	1	1	10″	9″	12″
h	Oil circulating pumps	Horizontal duplex	2	1	2″	2½″ oil.	2½″										
		Vertical simplex						4	1	4″	5″ oil.	6″	2	1	10″	9″ oil.	12″
		do															
i	Fire and bilge pumps	do	2	1	10″	8½″	12″	2	1	10″	8½″	12″	2	1	10″	9″	12″
k	Auxiliary air and circulating pumps.	Simplex horizontal combined.	1	1	6″	10″ / 10″ air.	12″	1	1	6″	10″ / 10″ air.	12″	1	1	6″	10″ / 10″ air.	12″
l	Auxiliary feed pumps	Vertical simplex	3	1	9″	6″	12″	3	1	9″	6″	12″	3	1	9″	6″	12″
m	F. and B. on ash ejector	do	3	1	12″	8″	12″	3	1	12″	8″	12″	3	1	12″	8″	12″
n	Forced draft blowers	Horizontal double	6	2	6″	66″ fan.	6″										
		Double upright											6	2	8″	84″ fan.	5″
		Horizontal double						6	2	6″	66″ fan.	6″					
				Horizontal.					Horizontal.					Vertical.			
o	Distiller circulating pumps	Simplex	2	1	10″	10″	12″	2	1	10″	10″	12″	2	1	10″	9″	12″
p	Evaporator feed pumps	Vertical simplex	1	1	4½″	5″	6″	1	1	4½″	5″	6″	1	1	4½″	5″	6″
r	Distiller fresh water pumps	do	1	1	4½″	5″	6″	1	1	4½″	5″	6″	1	1	4½″	5″	6″

TABLE S.—Continued.

Identification letter.	Auxiliaries.- Name.	Type.	Birmingham. No.	Diameter of cylinders. No.	Steam.	Water.	Stroke.	Salem. No.	Diameter of cylinders. No.	Steam.	Water.	Stroke.	Chester. No.	Diameter of cylinders. No.	Steam.	Water.	Stroke.
s	Air compressors	Westinghouse	2	2	11″	11″	12″	2	2	11″	11″	12″	2	2	11″	11″	12″
t	Dynamo plant	Cross compound	3	2	7½″ 12″		8″	3	2	7½″ 12″		8″	3	2	7½″ 12″		8″
u	Dynamo A. and C. pump	Simplex horizontal combined.	1	1	6″	8″ 8″ air.	12″	1	1	6″	8″ 8″ air.	12″	1	1	6″	8″ 8″ air.	12″
v	Ice machinery	1-ton Allen dense air	2	1	8″	Compressor, 6½″ Expander, 5½″.	8″	2	1	8″	Compressor, 6½″. Expander, 5″.	8″	2	1	8″	Compressor, 6½″. Expander, 5½″.	8″
w	Steering engine	Simple vertical	1	2	12″		10″	1	2	12″		10″	1	2	12″		10″
x	Anchor engine	do	1	2	12″		10″	1	2	12″		10″	1	2	10″-12″	Compound.	10″

TABLE 2

U.S.S. BIRMINGHAM
SEPTEMBER 29 - 1908.

TEST OF AFTER MAIN AIR PUMP
DIAM OF STEAM CYL 12"
" " WATER CYL 22" } BLAKE DOUBLE CYLINDER SIMPLEX PUMP,
" " PISTON ROD 2 7/8" WITH INDEPENDENT STEAM CYLINDERS.
STROKE 18"

	TIME	STEAM PRESSURE GAGE. BOILER	STEAM CHEST	VACUUM CONDENSER	AUX. CONDENSER AUX. PRESSURE IN AUX. EXH. LINE (GAGE)	REVOLUTIONS OR DOUBLE STROKES	MEAN PRESSURES FORWARD	MEAN PRESSURES AFTER	I.H.P.	CONDENSED EXHAUST STEAM: TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK LBS.	TEMPERATURE	WEIGHT PER HOUR LBS.	PER I.H.P.	DURATION OF TEST.
TEST NO 1	PM 12.00	210	50	22.7	5.3	268		16.	4.3	12.0000				84			
	12.10	235	50	22.7	6.	220		16.	3.6	12.2041	20.41	10	217	84			
	12.20	235	50	22.9	6.	261		16.5	4.6	12.4023	1.942	20	455	84			
	12.30	265	50	22.9	5.	257		15.	3.9	12.5805	17.40	30	688	84			
	12.40	234	50	22.9	5.	288		15.	4.4	1.1424	16.21	40	922	84			
	12.50	250	50	23.	6.	280		14.3	4.0	1.3021	15.57	FULL	1134	84			
	1.00	252	50	23.	5.	277		13.8	3.8	Vacuum in Main Condenser (inches of Mercury) 22.9					753.1	18.62.	1 Hr 30M. 21.5.
	1.10	250	50	23.	5.5	278		14.	3.9								
	1.20	245	50	23.	5.	277		13.	3.6								
	1.30	250	50	23.	5.	280		13.5	3.8								
	Average	243	50	22.9	5.4	269.	*	14.7	4.0								
TEST NO 2	2.40	250	44.	23.	5.	211		17.5	3.8	2.4000				86			
	2.50	250	43.	20.	5.	212		16.	3.4	2.5810	18.10	10	217	85			
	3.00	250	43.	23.	5.	216		16.	3.5	3.1812	20.02	20	455	84			
	3.10	250	43.	23.	5.	202		16.	3.3	3.3735	19.15	30	686	84			
	3.20	250	43.	23.	5.	195		17.	3.4	3.5642	19.07	40	922	82			
	3.30	250	45.	23.	5.	210		16.	3.4	4.1122	1.440	FULL	1134	82			
	3.40	250	45.	23.	5.	218		16.	3.5	Vacuum in Main Condenser (inches of Mercury) 23.0							
	3.50	250	45.	23.	5.	207		16.	3.4						744.8	21.2.8	1 Hr. 31 M. 22.5.
	4.00	250	45.	23.	4.8	219		16.	3.5								
	4.10	250	45.	23.	5.	220		16.5	3.6								
	Average	250	44.	23.	5.	211	*	16.3	3.5								
TEST NO 3	5.30	250	44.	21.	5.	142		19.8	2.8	5.3000				82			
	5.40	250	43.	21.	5.	144		19.5	2.8	5.5511	25.11	10	217	82			
	5.50	250	43.	21.	5.	143		19.6	2.9	6.2016	25.05	20	453	83			
	6.00	250	43.	21.	5.	142		19.	2.7	6.4507	24.51	30	688	83			
	6.10	250	42.	21.	5.	144		19.3	2.8	7.1552	26.45	40	922	83			
	6.20	250	43.	21.	5.	141		20.3	2.9	7.3940	25.48	FULL	1134	83			
	6.30	250	42.	21.	5.	144		19.6	2.8	Vacuum in Main Condenser (inches of Mercury) 21.0							
	6.40	250	42.	21.	5.	144		19.8	2.8								
	6.50	248	42.	21.	5.	142		20.3	2.9								
	7.00	250	42.	21.	5.	143		20.	2.9								
	7.10	250	42.	21.	5.	142		18.7	2.6						524.7	16.7.4	2 Hr. 3M. 40.5.
	7.20	250	42.	21.	5.	144		19.3	2.8								
	7.30	250	42.	21.	5.	143		19.8	2.9								
	7.40	250	42.	21.	5.	146		19.8	3.0								
	Average	250	42.	21.	5.	143	*	19.6	2.8								
TEST NO 4	8.05	250	43.	23.	5.	269		12.	6.5	8.0500				97			
	8.15	250	43.	23.	5.	266		13.	7.0	8.1226	726	10	216	97			
	8.25	250	43.	23.	5.	269		13.5	7.4	8.2105	8.37	20	451	96			
	8.35	250	43.	23.	5.	272		14.	7.6	8.2939	8.36	30	686	97			
	8.45	250	43.	23.	5.	271		14.	7.7	8.3816	8.37	40	919	96			
	Average	250	43.	23.	5.	269	§	13.3	§ 7.3	8.4548	7.30	FULL	1130	96			
										Vacuum in Main Condenser (inches of Mercury) 23.0					1663.1	22.7.8	0 Hr. 40M. 44.5.

*Forward cylinder not in operation during these tests.

§Both cylinders in operation but only one indicated. In calculations, each cylinder has been considered as developing equal power, and the total (both cylinders) power is entered in I.H.P. column.

TABLE 3

U.S.S. CHESTER.
OCTOBER 3 1908.

TEST OF AFTER MAIN AIR PUMP.

DIAM OF STEAM CYLS. 14"
WATER " 35"
PISTON RODS. 2¾"
STROKE, FULL 21"; on test 15½"

BLAKE VERTICAL TWIN BEAM.

	TIME	STEAM PRESSURE. LBS. BOILER	STEAM CHEST	VACUUM AUX. CONDENSER.	BACK PRESSURE IN AUX. EXH. LINE (GAGE)	REVOLUTIONS OR DOUBLE STROKES.	MEAN PRESSURE. PORT CYL.	MEAN PRESSURE. STBD CYL.	I.H.P.	CONDENSED EXHAUST STEAM. TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK LBS.	TEMPERATURE	PER HR. LBS. WEIGHT	PER HR. LBS. PER I.H.P.	DURATION OF TEST
TEST NO. 1	AM 10.30	130	112	9	6.					10:30:00	M. S.			84			
	10.40	130	110	9	6.	31.4	24.0	25.8	22.1	10:37:08	7.08	20	457	83			
	10.50	115	104	9	5.5	29.2	24.6	26.4	21.1	10:44:54	7.46	40	924	81			
	11.00	105	95	10	6.	26.0	26.0	27.0	19.6	10:47:29	2.35	FULL	1136	81			
	AVERAGE	120	105	9	5.9	28.9	24.9	26.4	21.0	10:57:45	10.16	30	688	83			
										11:05:39	7.54	FULL	1136	84			
TEST NO. 2	12.00	190	80	10	6.					12:00:00				75			
	12.10	205	80	10	6.5	21.8	25.4	29.3	16.9	12:05:46	5.46	10	219	75			
	12.20	200	82	10	6.5	21.7	25.4	29.4	16.8	12:11:39	5.53	20	457	74			
	12.30	205	85	10	6.	22.3	24.6	29.0	16.9	12:17:48	6.09	30	686	74			
	AVERAGE	200	82	10	6.3	21.9	25.1	29.2	16.9	12:23:55	6.07	40	924	75	3824.1	182.1	0 Hr. 28 M. 52.5 S.
										12:28:52	4.57	FULL	1136	75			
TEST NO. 3	PM 1.10	200	74	10	5.5	16.1	31.0	31.0	14.2	1:00:00				73			
	1.20	200	73	10	5.5	15.9	30.3	31.8	14.0	1:07:57	7.57	10	219	73			
	1.30	200	73	10	6.	15.9	30.3	32.8	14.3	1:16:39	8.42	20	457	73			
	1.40	200	73	10	5.5	15.7	30.3	32.3	13.9	1:25:40	8.01	30	689	73			
	AVERAGE	200	73	10	5.6	15.9	30.5	31.9	14.1	1:34:36	8.56	40	925	73	2361.0	139.7	0 Hr. 42 M. 49.5 S.
										1:42:49	8.13	FULL	1137	73			
TEST NO. 4	2.00	200	59	10	6.					2:00:00				72			
	2.10	200	59	10	6.	9.5	27.8	30.8	7.9	2:14:31	14.31	10	219	72			
	2.20	200	59	10	5.5	9.6	28.3	30.8	8.0	2:29:43	15.12	20	457	72			
	2.30	200	60	10	6.	9.6	26.8	30.0	7.8	2:45:20	15.37	30	689	72			
	2.40	200	58	10	6.	9.6	27.3	30.0	7.8	3:01:31	16.07	40	925	72			
	2.50	200	58	10	6.	9.7	27.3	30.3	7.9	3:15:45	13.02	FULL	1137	72			
	3.00	200	58	10	5.5	9.6	27.3	29.7	7.8								
	3.10	200	58	10	6.	9.6	27.3	29.6	7.8								
	3.20	200	58	10	5.5	9.7	26.5	30.0	7.8								
	AVERAGE	200	58	10	5.8	9.6	27.3	30.2	7.8						300.9	115.5	1 Hr. 15 M. 43 S.

	Test 1	Test 2	Test 3	Test 4
Vacuum in Main Condenser (inches of Mercury)	23.1	21	18.1	14

TABLE 4

U. S. S. SALEM.
OCTOBER 9 - 1908

TEST OF AFTER TURBINE WET VACUUM PUMP
VERTICAL CENTRIFUGAL 2-STAGE HOT WELL PUMP
DRIVEN BY A 27" 2-STAGE CURTIS TURBINE.

	TIME	STEAM PRESSURES BOILER	STEAM CHEST	1ST. STAGE LBS	2ND. STAGE INS.	VACUUM AUX. CONDENSER.	BACK PRESSURE IN AUX. EXH. LINE (GAGE)	REVS. / 2.4	REVOLUTIONS	CONDENSED EXHAUST STEAM: TIME OF OBSERVATION	INTERVAL	INCHES IN TANK.	WEIGHT IN TANK LBS	TEMPERATURE	PER HOUR LBS: WEIGHT	PER I.H.P.	DURATION OF TEST
TEST NO. 1	4.40	250	180	5	16	20	15	516	1238	4.40.00	M. S.			80			
	4.50	250	220	5	16	15	15	555	1332	5.02.05	22.05	10	206	82			
	5.00	250	200	4	15	19	15	545	1308	5.29.53	27.48	20	441	80			
	5.10	•	200	5	15			547	1313	5.58.15	28.22	30	677	81			
	5.20	248	190	5	16	20	15	557	1337	6.24.55	26.40	40	919	81			
	5.30	250	185	5	14	20	15	555	1332	6.53.40	28.45	FULL	1132	81			
	5.40	250	190	6	15	20	15	533	1279								
	5.50		190	4	14			544	1306								
	6.00	250	225	10	15	20	15	546	1310	One nozzle to Steam Turbine open.							
	6.10	250	190	4	15	20	15	555	1332								
	6.20	250	220	5	16	20	15	556	1335								
	6.30		195	3	17			538	1315								
	6.40	250	200	4	19	20	16	550	1320								
	6.50	250	205	4	19	20	17	556	1335						508.1		2 Hr. 13 M. 40 S.
	AVERAGE	249	199	5	16	19	15	547	1314								
TEST NO. 2	7.20	250	160	22	5	10	5	540	1296	7.20.00				81			
	7.30	245	135	20	4	10	6	547	1313	7.34.58	16.58	10	206	81			
	7.40	250	160	20	7	10	5	542	1301	7.55.30	20.32	20	441	79			
	7.50	240	155	25	6	10	6	541	1298	8.16.25	20.55	30	677	78			
	8.00	245	160	25	6	10	6	532	1277	8.34.28	18.03	40	919	77			
	8.10		180	25	7	10	6	534	1282	8.55.21	19.53	FULL	1132	77			
	8.20	245	145	20	6	10	6	546	1310								
	8.30	250	148	20	5	10	6	545	1308	Two nozzles to Steam Turbine open.							
	8.40	250	145	25	6	11	5	520	1248								
	8.50	248	135	20	5	11	6	544	1306								
	9.00	246	130	20	6	11	6	530	1272						712.5		1 Hr. 35 M. 21 S.
	AVERAGE	247	150	22	6	10	6	539	1292								

TABLE 5

U.S.S. SALEM
OCTOBER 9 1908.

TEST OF AFTER DRY VACUUM PUMP.

DIAM OF STEAM CYL 10"
" WATER " 24"
" PISTON ROD 2⅜"
STROKE 18"

	TIME	STEAM PRESSURE GAGE: BOILER	STEAM PRESSURE GAGE: STEAM CHEST	VACUUM. AUX. CONDENSER.	BACK PRESSURE IN AUX. EXH. LINE. (GAGE)	VACUUM.	REVOLUTIONS.	MEAN PRESSURE: CYLINDER	MEAN PRESSURE: I.H.P.	CONDENSED EXHAUST STEAM: TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK LBS.	TEMPERATURE.	PER HOUR LBS.: WEIGHT	PER HOUR LBS.: PER I.H.P.	DURATION OF TEST.
TEST NO 1	M 10.45	240	85	10	6.	19.		493		10.45.00	M. S.			F 74			
	10.55	240	83	10	6.	193	60.8	50.8	215	1.05.08	8.08	10	206	74			
	11.05	240	85	10	6.	193	59.7	47.7	198	11.04.10	10.62	20	442	74			
	11.15	240	84	10	6.	19.	58.8	48.8	200	11.15.04	10.44	30	678	74			
	11.25	240	87	10	6.	193	60.2	49.	206	11.25.03	9.59	40	921	73			
	11.35	240	88	10	6.	193	61.0	50.3	214	11.34.30	9.27	FULL	1133	73	1373.3	66.5	Dhr 43M 30 S.
	Average	240	85	10	6.	192	60.1	49.3	207								
TEST NO 2	1.25	236	75	10	6.	213				1.25.00				78			
	1.35	241	70	10	6.	218	77.6	46.4	25.1	1.32.45	7.45	10	206	78			
	1.45	241	70	10	6.	213	83.2	46.1	26.7	1.40.55	8.10	20	442	78			
	1.55	242	70	10	6.	215	80.4	46.1	25.8	1.49.26	8.41	30	678	77			
	2.05	242	70	10	6.	215	79.0	46.7	26.8	1.58.18	7.52	40	921	77			
	Average	240	71	10	6.	215	80.1	46.8	26.1	2.07.05	7.45	FULL	1133	76	1615.4	61.89	Dhr 42M 05S.
TEST NO 3	2.15	241	65	10	5.	18.				2.15.00				74			
	2.25	242	65	10	5.5	173	44.8	46.5	14.5	2.26.47	11.47	10	206	72			
	2.35	241	67	10	5.	175	45.2	46.	14.5	2.38.07	11.20	20	442	72			
	2.45	240	70	10	6.	175	43.6	44.8	13.6	2.51.25	13.18	30	678	72			
	2.55	240	70	10	6.	178	45.6	44.1	14.0	3.03.25	12.00	40	921	71			
	3.05	240	75	10	6.	178	45.9	44.8	14.3	3.14.48	11.23	FULL	1133	71			
	3.15	241	76	10	6.	178	46.1	44.4	14.2						1136.8	80.0	Dhr 59 M. 48 S.
	Average	241	70	10	5.6	177	45.2	45.1	14.2								

TABLE 6

U. S. S. BIRMINGHAM
SEPTEMBER 29-1900

TEST OF AFTER MAIN CIRCULATING PUMP ENGINE (COMPOUND)

DIAM OF H.P CYL 8"
L.P 16"
PISTON RODS 1 7/16"
STROKE 8"

	TIME	STEAM PRESSURES, GAGE: BOILER	FORD CHEST	AFT CHEST	VALVE IN AUX. CONDENSER	BACK PRESSURE IN AUX. EXH. LINE (GAGE)	DISCHARGE PRESSURE	REVOLUTIONS OR DOUBLE STROKES	MEAN PRESSURES: H.P. CYL	L.P. CYL	I.H.P. TOTAL	CONDENSED EXHAUST STEAM: TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK LBS	TEMPERATURE	PER HOUR LBS. WEIGHT	PER I.H.P.	DURATION OF TEST
TEST NO 1	PM 1.15	245		105	30	5	10	232	603	111	485	1:1500	M. S			*92			
	125	250		105	30	5	10	244	613	103	493	1:2220	720	10	216	92			
	135	250		105	30	5	10	244	598	111	508	1:3107	847	20	451	92			
	145	250		105	30	5	10	240	603	105	490	1:3959	852	30	686	92			
	155	250		102	30	5	10	236	578	106	472	1:4835	836	40	919	92			
	AVERAGE	249		1044	30	5	10	239	599	107	489	1:5625	750	FULL	1130	92	1631.0	33.5	0 Hr 41 M 25 S.
TEST NO 2	210	250		65	30	5	8	174	397	50	207	2:1000				*93			
	220	250		60	30	5	8	172	398	49	204	2:2646	1646	10	216	90			
	230	250		65	30	5	8	176	418	53	221	2:4336	1650	20	453	88			
	240	250		58	30	5	8	174	383	54	208	2:5854	1518	30	688	88			
	250	250		58	30	5	8	178	398	55	219	3:1349	1455	40	922	86			
	300	250		58	30	5	8	174	383	56	211	3:2833	1444	FULL	1134	86			
	310	250		60	30	5	8	180	408	57	228								
	320	250		52	30	5	8	178	398	59	225								
	330	250		58	30	5	8	176	395	55	216						866.2	40.3	1 Hr 18 M 33 S.
	AVERAGE	250		593	30	5	8	177	397	54	215								
TEST NO 3	350	250		32	30	5	6	112	218	16	63	3:5000				88			
	400	250		30	30	5	6	112	220	17	64	4:1440	2450	10	217	86			
	410	250		33	30	5	6	112	220	19	66	4:4731	3291	20	453	86			
	420	250		32	30	5	6	110	223	19	66	5:2450	3719	30	688	86			
	430	250		31	30	5	6	110	218	17	63	6:0143	3655	40	922	86			
	440	250		30	30	5	6	110	213	20	64	6:3315	3152	FULL	1134	86			
	450	250		32	30	5	6	106	210	15	57								
	500	250		33	30	5	6	108	213	16	60								
	510	250		33	30	5	6	112	208	14	59								
	520	250	(recorded)	32	30	5	6	108	210	14	57								
	530	250		32	30	5	6	110	183	20	58								
	540	250		33	30	5	6	110	208	17	60								
	550	250		33	30	5	6	112	210	11	57								
	600	250	Not	33	30	5	6	108	210	13	56								
	610	250		33	30	5	6	110	210	13	58								
	620	250		33	30	5	6	110	213										
	630	250		33	30	5	6	110	213								416.7	69.4	2 Hr 25 M 15 S.
	AVERAGE	250		321	30	5	6	110	212	16	60								
TEST NO 4	650	250		14	30	5	8	64	126	10	21	6:5000				84			
	710	250		14	30	5	8	57	113	10	17	7:0534	1534	5	97	84			
	730	250		14	30	5	8	56	110	08	16	7:3035	2501	10	217	84			
	750	250		14	30	5	8	56	110	09	16	8:1838	4803	15	336	86			
	810	250		17	30	5	8	56	113	10	17	8:5455	3615	20	452	90			
	830	250		15	30	5	8	52	110	08	14	9:2611	3118	25	571	88			
	850	250		22	30	5	8	66	144	10	24								
	910	250		17	30	5	8	58	129	09	19								
	930	250		15	30	5	8	58	119	07	17								
	AVERAGE	250		157	30	5	8	58	119	9	18						219.3	12.18	2 Hr 36 M 0 S.

NOTE.—Main injection and discharge valves wide open during all tests.

TABLE 7

TEST OF FORWARD MAIN CIRCULATING PUMP

U.S.S. SALEM.

OCT. 6 1908.

(TWO CYLINDERS COMPOUND)

DIAM OF H.P CYLINDER 8"
" " L.P. " 16"
" " PISTON RODS 1¾"
STROKE 8"

	TIME	STEAM PRESSURE GAGE: BOILER	H.P. CHEST	L.P. CHEST	VACUUM AUX. CONDENSER	BACK PRESSURE IN AUX. EXH. LINE (GAGE)	DISCHARGE PRESSURE	REVOLUTIONS OR DOUBLE STROKES	MEAN PRESSURE: H.P. CYL.	L.P. CYL.	I.H.P. TOTAL.	CONDENSED EXHAUST STEAM: TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK. LBS.	TEMPERATURE	PER HOUR LBS.: WEIGHT	PER I.H.P	DURATION OF TEST
TEST NO 1	PM 3.15	240	50	12	10	6						3.15.00				64			
	3.25	240	50	10	10			870	10.5	2.4	35	3.31.17	16.17	5	86	83			
	3.35	240	48	11	10	6		865	10.0	2.5	35	3.56.52	25.35	10	206	80			
	3.45	240	47	10	10			868	10.5	2.1	33	4.20.50	23.58	15	325	79			
	3.55	240	46	11	10	5.5		857	10.5	2.5	35	4.42.33	21.43	20	439	79			
	4.05	240	45	11	10			850	10.0	2.3	33	5.07.14	24.41	25	546	79			
	4.15	240	45	12	10	6		893	10.5	2.9	39								
	4.25	240	46	12	10			878	9.9	2.6	36								
	4.35	240	45	11	10			892	10.8	2.7	39								
	4.45	240	45	12	10	5.5		904	9.8	3.0	39								
	4.55	240	45	11	10			876	10.1	2.0	32								
	5.05	240	45	11	10	6		872	11.8	2.6	39								
	AVERAGE	240	46	11	10	6		875	10.4	2.5	36						302.6	84.1	1 Hr. 52 M. 14 S.
TEST NO 2	6.50	246	81	17	10	6						6.50.00				71			
	7.00	240	86	18	10	6		1580	22.8	6.5	154	6.56.10	6.10	5	86	71			
	7.10	241	87	18	10	6	taken.	1544				7.06.03	9.53	10	206	71			
	7.20	240	80	17	10	6.5		1533	22.4	6.9	154	7.15.23	9.20	15	325	71			
	7.30	240	77	16	10	6		1543	23.1	6.4	150	7.24.25	9.02	20	439	71			
	7.40	242	75	16	10	6		1470	22.3	7.0	148	7.32.50	8.25	25	566	71			
	7.50	243	82	17	10	6	Not	1574	25.3	7.1	165	7.41.50	9.00	30	673	71			
	8.00	245	81	18	10	5		1577	25.1	7.7	177	7.50.20	8.30	35	787	71			
	8.10	243	88	18	10	5.5		1590	25	7.2	171	7.58.10	7.50	40	906	71			
	AVERAGE	242	82	17	10	6		1551	23.7	6.9	159	8.07.07	8.57	45	1023	71			
												8.14.33	7.26	FULL	1121	71	795.5	50.0	1 Hr. 24 M. 33 S.
TEST NO 3	8.40	240	180	44	10	6						8.40.00							
	8.50	250	170	43	10	6		2754	64	21.5	827	8.42.20	2.20	5	86	98			
	9.00	240	170	42	10	7		2762	63	20.1	793	8.44.55	2.36	10	205	98			
	AVERAGE	243	173	43	10	6.7		2758	63.5	20.8	810	8.47.50	2.55	15	324	98			
												8.49.40	1.50	20	437	100			
												8.52.30	2.50	25	564	104			
												8.55.15	2.45	30	670	104			
												8.57.45	2.30	36	784	105			
												9.00.18	2.33	40	902	105			
												9.03.02	2.44	45	1019	105			
												9.06.20	2.18	FULL	1116	105	2643.0	32.6	0 Hr. 25 M. 20 S.
TEST NO. 4	9.15	230	200	55	10	6						9.15.00				148			
	9.25	235	200	55	10	6		3979	86.8	29.3	1626			5	85	148			
	9.35	245	180	49	10	6.5		4145	75.3	29.5	1606	9.19.00		10	202	148			
	AVERAGE	236	193	53	10	6.2		4062	81.0	29.4	1616	9.22.15	1.15	15	310	148			
												9.23.57	1.32	20	431	145			
												9.25.15	1.18	25	551	150			
												9.26.52	1.37	30	661	150			
												9.28.35	1.43	35	774	150			
												9.28.20	1.45	40	890	144			
												9.30.10	1.50	45	1006	140			
												9.31.37	1.27	FULL	1105	138	3990.	247	0 Hr. 16 M. 37 S.

NOTE.—Main injection and discharge valves wide open during all tests.

TABLE 8 U.S.S. CHESTER
OCTOBER 2 1908.

TEST OF AFTER MAIN CIRCULATING PUMP. (TWO CYLINDER COMPOUND).

DIAM. OF H.P. CYLINDER 10
" " L.P. " 16
" " PISTON ROD 1 5/8"
STROKE 10"

	TIME	STEAM PRESSURE. GAGE. BOILER.	H.P. CHEST.	L.P. CHEST.	VACUUM. AUX. CONDENSER.	BACK PRESSURE IN AUX. EXH. LINE. (GAGE.)	DISCHARGE PRESSURE.	REVOLUTIONS. OR DOUBLE STROKES.	MEAN PRESSURE H.P. CYL.	MEAN PRESSURE L.P. CYL.	I.H.P. (TOTAL)	CONDENSED EXHAUST STEAM. TIME OF OBSERVATION	INTERVAL.	INCHES IN TANK.	WT. IN TANK. LBS.	TEMPERATURE.	PER HOUR LBS. WEIGHT.	PER I.H.P.	DURATION OF TEST.
TEST NO. 1	PM. 4.30	200	120	50	10	6.						4.30:00	M. S.			F 80			
	4.40	210	125	50	10	5.5		2309	76.5	33.1	145.6	4.42:11	12.11	30	688	79			
	4.50	210	120	50	10	5.5		2391	74.6	33.0	154.0	4.49:51	7.20	FULL	1136	78			
	5.00	215	125	50	10	5.5		2307	79.3	29.0	140.	4.57:16	7.45	20	456	79			
	5.10	198	150	50	10	6.		2320	75.0	27.7	132.2	5.05:19	8.03	40	927	80			
	AVERAGE	206	128	50	10	5.7		2332	76.4	30.9	142.9	5.09:12	3.53	FULL	1139	80	3472	24.3	0HR 09M 45S. 0HR 19M 31S.
TEST NO. 2	5.30		100	35	10							5.30:00				75			
	5.40	175	100	35	10	6.		1818	67.	20.2	84.2	5.37:20	7.20	10	219	75			
	5.50	180	100	35	10	6.		1887	67.	20.5	88.0	5.44:48	7.28	20	456	76			
	6.00	175	100	35	10	7.		1903	70.8	19.3	89.2	5.52:15	7.27	30	688	76			
	AVERAGE	177	100	35	10	6.		1869	68.3	20.0	87.0	5.59:05	6.50	40	924	76			
												6.05:28	6.23	FULL	1136	76	1922	22.0	0 Hr. 35M. 28S.
TEST NO. 3.	6.20	180	75	25.	10	6.5						6.20:00				75			
	6.30	170	70	25.	10	6.		1453	35.6	10.6	35.6	6.31:35	11.35	10	219	75			
	6.40	170	70	22.5	10	6.		1454	36.0	10.3	35.4	6.44:28	12.53	20	456	76			
	6.50	195	75	22.5	10	6.		1434	25.4	19.4	42.1	6.55:53	11.25	30	688	76			
	7.00	180	75	25.	10	5.5		1532	35.0	10.5	37.0	7.08:41	12.48	40	924	76			
	7.10	170	75	25.	10	6.5		1412	40.0	11.0	38.8	7.20:20	11.39	FULL	1136	76			
	7.20	152	70	22.5	10	5.5		1438	35.0	9.3	33.0								
	AVERAGE	174	73	23.9	10	6.0		1450	34.5	11.8	37.0						1135	30.6	1 Hr. 0M. 20 S.
TEST NO. 4	7.45	230	35	12.5	10	7.						7.45:00				77			
	7.55	260	35	12.5	10	6.		81.7	15.4	4.7	8.7	8.14:15	29.15	10	219	76			
	8.05	240	33	12.5	10	6.		788				8.49:58	35.43	20	456	80			
	8.15	225	35	12.5	10	6		762	14.3	4.5	7.7	9.07:32	17.34	25	573	81			
	8.25	235	35	12.5	10	6.	Not taken.	761				9.25:37	18.05	30	688	81			
	8.35	235	35	12.5	10	6		763	14.5	7.0	9.7	9.42:56	17.18	35	806	82			
	8.45	235	35	12.5	10	6.		766											
	8.55	235	35	12.5	10	6.5		754	15.0	7.0	9.7								
	9.05	225	35	12.5	10	6.		774											
	9.15	230	35	12.5	10	6.		766	13.8	6.5	9.1								
	9.25	235	33	12.5	10	5.5		752											
	9.35	230	35	12.5	10	6.		758	14.5	8.5	10.8								
	9.45	225	35	12.5	10	6.		760											
	AVERAGE	234	35	12.5	10	6.8		768	14.6	6.4	9.3						411.1	44.2	1 Hr. 57 M. 55 S.

NOTE.—Main injection and discharge valves wide open during all tests.

TABLE 9

U.S.S. BIRMINGHAM
OCTOBER 12-1908.

TEST OF MAIN FEED PUMP IN AFTER ENGINE ROOM

DIAM. OF STEAM CYL. 15"
" " WATER " 10"
" " PISTON ROD 2 3/16"
STROKE 15"

	TIME	STEAM PRESSURE GAGE. BOILER	STEAM CHEST	VACUUM AUX. CONDENSER.	BACK PRESSURE IN AUX. EXH LINE (GAGE)	DISCHARGE PRESSURE.	DOUBLE STROKES	MEAN PRESSURE CYLINDER	I.H.P.	CONDENSED EXHAUST STEAM. TIME OF OBSERVATION	INTERVAL	INCHES IN TANK.	WEIGHT IN TANK. LBS.	TEMPERATURE.	PER HOUR LBS. WEIGHT	PER I.H.P.	DURATION OF TEST.
TEST NO 1	P.M. 6.05	195	200	26	6	200	50	1068	703	6.05:00	M. S.			114			
	6.15	195	195	26	6	200	50	858	565	6.07:52	2.52	10	214	114			
	6.20	210	200	26	7	200	50	110	724	6.10:45	2.53	20	446	114			
	AVERAGE	200	198	26	63	200	50	1008	664	6.13:39	2.54	30	682	114			
										6.16:34	2.55	40	914	114			
										6.19:32	2.58	FULL	1129	114			
TEST NO. 2	6.43	225	195	26	6	200	32	116	489	6.43:00				96			
	6.48	225	190	26	6	200	30	120	474	6.46:44	3.44	10	215	96			
	6.53	225	185	26	6	195	30	116	458	6.50:58	4.14	20	448	96			
	6.58	225	175	26	6	225	30	114	450	6.55:37	4.39	30	685	96			
	7.03	225	175	26	6	225	30	115	454	7.00:12	4.35	40	918	96			
	7.08	225	175	26	6	225	30	116	458	7.04:10	3.58	FULL	1134	96			
	AVERAGE	225	183	26	6	212	30	1161	464								

4661.0 / 702 — 0 Hr. 14 M. 32.5

3214.5 / 693 — 0 Hr. 21 M. 10.5

TABLE 10

U.S.S. SALEM
OCTOBER 7-1908.

TEST OF (AFTER) MAIN FEED PUMP (BLAKE SIMPLEX)

DIAMETER STEAM CYLINDER 15"
DIAMETER OF PISTON ROD 2½"
DIAMETER WATER CYLINDER 10"
STROKE 15"

	TIME	STEAM PRESSURES GAGE. BOILER	STEAM CHEST OF PUMP.	VACUUM AUX. CONDENSER. BACK PRESSURE IN AUX. EXH LINE (GAGE)	DOUBLE STROKES OF PUMP.	MEAN PRESSURE CYLINDER	I.H.P.	CONDENSED EXHAUST STEAM. TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK LBS	TEMPERATURE	WEIGHT. PER HOUR LBS.	PER I.H.P.	DURATION OF TEST.
TEST NO 1	P.M. 2.20	210	160	10	6.			2:20:00	H. &			92			
	2.30	250	165	10	7.	108	119.0	2:30:05	10.05	10	205	90			
	2.40	250	170	10	5.	133	118.8	2:36:20	6.15	20	438	89			
	2.50	248	165	10	6.	135	112.9	2:43:04	6.44	30	671	88			
	3.00	245	165	10	5.5	137	117.5	2:50:06	7.02	40	904	88			
	AVERAGE	241	165	10	5.9	128	117.0	2:57:47	7.41	FULL	1118	87	1775.3	89.7	0 HR. 37 M. 41 S.
TEST NO 2	3.20	235	200	10	7.			3:20:00				112			
	3.25	230	200	10	6.5	24.	128.8	3:24:31	4.51	10	205	112			
	3.30	240	200	10	5.5	232	124.9	3:29:37	5.06	20	437	108			
	3.35	250	210	10	6.5	252	130.0	3:34:15	4.38	30	670	107			
	3.40	245	205	10	5.5	258	128.8	3:38:35	4.20	40	902	107			
	3.45	245	205	10	6.	256	127.0	3:42:57	4.22	FULL	1116	107	2917.0	70.0	0 HR. 22 M. 57 S.
	AVERAGE	241	203	10	6.2	248	127.9								
TEST NO 3	4.15	230	215	10	7.			4:15:00				114			
	4.20	230	218	10	6.	34.	123.0	4:19:21	4.21	10	204	114			
	4.25	230	220	10	6.	35.	125.2	4:22:01	3.40	20	435	115			
	4.30	220	215	10	5.	356	120.0	4:25:35	3.34	30	667	116			
	4.35	205	185	10	7.	338	111.3	4:29:20	3.45	40	898	117			
	4.40	200	185	10	5	312	112.2	4:33:28	4.08	FULL	1111	118			
	4.45	205	205	10	6.	278	117.0	4:41:00				115			
	4.50	220	220	10	6.	344	118.2	4:48:08	2.08	10	204	115			
	4.55	215	225	10	6.	336	108.0	4:45:57	2.49	20	435	116			
	5.00		220	10		374	118.0	4:48:00	3.03	30	667	118			
	AVERAGE	218	211	10	6.2	345	117.0	4:51:58	2.58	40	898	121	4185.6	79.4	0 HR. 40 M. 0 S.
	10.05	250	245	11	6.			4:55:00	3.02	FULL	1111	121			
TEST NO 4	10.15	255	245	11	6.	454	124.0	10:05:00				154			
	10.25	260	245	11	6.	459	119.3	10:07:21	2.21	10	203	134			
	10.35	240	225	11	6.	447	119.3	10:10:12	2.51	20	430	158			
	AVERAGE	251	240	11	6	453	120.9	10:12:42	2.30	30	660	154			
								10:14:56	2.14	40	886	155			
								10:17:17	2.21	FULL	1099	154			
								10:17:17				154			
								10:19:50	2.33	10	204	116			
								10:21:39	1.49	20	439	114			
								10:24:15	2.36	30	673	115			
								10:26:50	2.35	40	915	112			
								10:29:30	2.40	FULL	1125	111	5446.7	74.55	0 HR. 24 M. 30 S.
TEST NO 5	10.45	250	220	10	6.			10:45:00				95			
	10.55	270	225	10	6.	336	102.0	10:48:25	3.25	10	205	95			
	11.05	250	220	10	6.	349	129.8	10:52:05	3.40	20	440	95			
	AVERAGE	257	223	10	6.	343	115.9	10:55:28	3.23	30	675	96			
								10:58:40	3.12	40	917	96			
								11:02:12	3.32	FULL	1128	96	3934.9	75.1	0 HR. 17 M. 12 S.

NOTE.—Average discharge pressure of pump in all the tests was about 300 (gauge) pounds. There were, however, violent fluctuations of the gauge pointer, the limits usually being about 275 as a minimum and 325 as a maximum.

TABLE II

U.S.S. CHESTER
OCTOBER 6 - 1908.

TEST OF MAIN FEED PUMP. (AFTER)

DIAM. OF STEAM CYL. 15"
" " WATER " 10"
" " PISTON RODS 2¾"
STROKE 15"

	TIME	STEAM PRESSURES. GAGE		VACUUM AUX. CONDENSER.	BACK PRESSURE IN AUX. EXH. LINE. (GAGE)	DISCHARGE PRESSURE.	DOUBLE STROKES	MEAN PRESSURE		CONDENSED EXHAUST STEAM.							
		BOILER	STEAM CHEST					CYLINDER	I.H.P.	TIME OF OBSERVATION	INTERVAL	INCHES IN TANK.	WEIGHT IN TANK. LBS.	TEMPERATURE.	PER HR LBS. WEIGHT	PER I.H.P.	DURATION OF TEST
TEST NO 1	P.M. 1.45	220	160	20	6.	280				P.M. 4.00	M. S.			° 76			
	1.55	220	158	20	6.	280	12.5	110.3	18.1	1:50:14	5.14	10	219	76			
	2.05	205	145	20	6.	250	12.5	112.	18.4	1:56:09	5.55	20	457	76			
	2.15	190	150	16	6	260	14.0	97.	17.9	2:01:57	5.48	30	688	76			
	Average	209	153	19	6.	268	13.	106.4	18.1	2:07:52	5.55	40	924	76			
										2:13:37	5.45	FULL	1136	76	2381.7		0H 25M. 37.5
TEST NO 2	2.30	200	185	20	6.	260				2:30:00				95			
	2.40	200	185	25	5.	265	25.	131.2	43.2	2:33:34	3.34	10	216	95			
	2.50	215	190	25	6.	270	24.	127.2	40.2	2:38:32	4.58	20	455	95			
	Average	205	187	23	5.7	265	24.5	129.2	41.7	2:42:29	3.57	30	686	95			
										2:47:18	4.49	40	921	95			
										2:51:02	3.44	FULL	1132	94	3229.4		0H 21M 02.5
TEST NO 3	3.10	270	255		5.	280	34.			3:10:00				99			
	3.20	275	255		6.	290	33.	138.2	60.4	3:17:55	7.55	30	686	99			
	3.30	265	255		6.	290	33.	141.	61.2	3:23:29	5.34	FULL	1132	99			
	3.40	270	250		5.	285	32	135.5	57.1	3:26:30	3.01	10	218	99			
	Average	270	254		5.5	286	33.	138.2	59.6	3:32:39	6.09	30	686	99	4766.4		
										3:38:30	5.51	FULL	1132	99		80.0	0H 20M 30.5

TABLE 12

U.S.S. BIRMINGHAM
OCTOBER 10 - 1908.

TEST OF AFTER ENGINE ROOM FIRE AND BILGE PUMP ENGINE.

DIAM. OF STEAM CYL. 10"
" " WATER " 8½"
" " PISTON RODS. 1⅝"
STROKE 12"

	TIME	STEAM PRESSURES GAGE		VACUUM	AUX. CONDENSER. AUX. PRESSURE IN AUX. EXH. LINE (GAGE)		DISCHARGE PRESSURE	REVOLUTIONS OR DOUBLE STROKES	MEAN PRESSURE		CONDENSED EXHAUST STEAM.							
		BOILER	STEAM CHEST						CYLINDER	I.H.P.	TIME OF OBSERVATION	INTERVAL	INCHES IN TANK.	WEIGHT IN TANK LBS.	TEMPERATURE	PER HOUR LBS. WEIGHT	PER I.H.P.	DURATION OF TEST.
TEST NO 1	P.M. 5.40		65				30	55.	37.5	97	6:10:00				° 70			
	5.50		65				30	55.	35.9	93	6:25:02	15.02	10	216	70			
	6.00		65				30	55.	35.4	91	6:41:04	16.02	20	450	70			
	6.10	140	100	3.	7.		30	55.	36.	93	6:56:52	15.46	30	687	69			
	6.20	120	65	3.	6.		30	55.	33.3	86	7:09:50	13.00	40	921	69			
	6.30	130	70	3.	6.		30	55.	34.8	90	7:18:20	8.30	FULL	1138	69			
	6.40	125	75	3.	5.5		30	55.	39.5	102								
	6.50	125	75	3.	6.		30	55.	37.8	97								
	7.00	145	80	3.	7.		30	55.	38.9	100								
	7.10	155	100	3.	7.		30	55.	41.9	108								
	7.20	180	65	3.	5.		30	55.	36.6	94								
	Average	140	75	3.	6.2		30	55.	37.1	95						999.2		1Hr. 8 M. 20.5.
TEST NO 2	7.50	150	100	3.	8.		45	50.	41.8	98	7:50:00				70			
	8.00	140	100	3.	7.		45	54.	45.7	116	7:54:55	4.55	5	97	70			
	8.10	120	100	3.	7.		45	54.	45.	114	8:00:41	5.46	10	216	70			
	8.20	120	95	3.	7.		45	54.	45.	114	8:06:45	6.04	15	334	70			
	8.30	120	100	3.	7.		45	54.	44.5	113	8:12:42	5.57	20	450	70			
	Average	130	99	3.	7.2		45	53.2	44.4	111	8:19:35	6.53	25	569	70			
											8:25:55	6.20	30	687	70	1147.7		0Hr. 35M. 55.5.
TEST NO 3	8.40	140	50	3.	6.		15	54.	22.8	58	8:40:00				70			
	8.50	150	50	3.	5.		15.	54.	22.6	57	8:48:15	8.15	5	97	70			
	9.00	155	45	3	5.5		15	54.	21.8	55	8:59:08	10.53	10	216	70			
	9.10	150	45	3.	6.		15	54.	20.6	52	9:11:18	12.10	15	334	70			
	9.20		45	3.			15	54.	22.5	57	9:24:43	13.25	20	450	70			
	9.30	150	45	3.	6.		15	54.	21.3	54	9:37:35	13.52	25	569	69			
	9.40	150	45	3.	5.5		15	54.	24.3	62	9:50:56	13.21	30	687	69			
	9.50	145	45	3.	6.		15	54.	22.0	56								
	Average	148	46	3.	5.7		15	54.	22.2	56						581.1		
																	103.9	1 Hr. 10M. 56.5

TABLE 13

U. S. S. SALEM.
OCTOBER 7-1908

TEST OF AFTER ENGINE ROOM FIRE AND BILGE PUMP ENGINE

DIAM OF STEAM CYL. 10"
WATER 8½"
PISTON RODS. 1¾"
STROKE 12"

	TIME	STEAM PRESSURES, GAGE. BOILER	STEAM CHEST	VACUUM AUX. CONDENSER. BACK PRESSURE IN AUX. EXH. LINE (GAGE)		DISCHARGE PRESSURE	REVOLUTIONS OR DOUBLE STROKES	MEAN PRESSURE CYLINDER	I.H.P.
TEST NO 4	PM 6.45	242	45	10	5.	10			
	6.55		48			10	395	13.4	2.5
	7.05	245	48	10	5.	10	407	14.3	2.7
	7.15		48			10	389	11.9	2.2
	7.25	240	50	10	6.	10	426	12.	2.4
	7.35		50			10	400	15.	2.8
	7.45	240	52	10	6	10	408	13.3	2.5
	7.55	242	53	10	6.	10	481	13.	3.0
	8.05		50			10	473	14.	3.1
	8.15	230	46	10	5.5	10	353	10.5	1.7
	8.25	240	50	10	6	10	457	12.9	2.8
	8.35								
	AVERAGE	240	49	10	5.6	10	419	13.0	2.6
TEST NO 2	8.45	240	50	10	7	20			
	8.55	237	55	10	7.	20	374	22.8	4.
	9.05	240	55	10	7	20	382	21.1	3.8
	9.15	242	57	10	7.	20	372	20.0	3.5
	9.25	240	50	10	5	20	419	21.4	4.2
	9.35		60			20	392	16.9	3.1
	9.45	244	60	10	6.	20	410	21.5	4.1
	9.55	240	60	10	6.	20	387	19.5	3.5
	10.05	240	55	10	7.	20	409	21.0	4.0
	10.15	241	55	10	7	20	370	19.0	3.3
	10.25	243	55	10	7.	20	434	20.8	4.2
	10.35	241	62	10	6.5	20	419		
	AVERAGE	241	56	10	6.6	20	397	20.4	3.8
TEST NO 3	10.45	242	85	10	5.	40			
	10.55	240	87	10	6.	40	413	36.1	7.
	11.05	242	86	10	6.	40	468	35.9	7.9
	11.15	240	90	10	7.	40	400	36.	6.8
	11.25	240	92	10	6	40	394	36.5	6.7
	11.35	244	85	10	6	40	404	37.3	7.1
	11.45	240	87	10	6	40	410	38.4	7.4
	11.55	230	85	10	5	40	360	38.3	6.5
	12.05	250	85	10	6.	40	372	39.5	6.9
	12.15								
	AVERAGE	241	87	10	5.9	40	403	37.3	7.0

CONDENSED EXHAUST STEAM.							
TIME OF OBSERVATION	INTERVAL. M. S.	INCHES IN TANK	WEIGHT IN TANK. LBS	TEMPERATURE.	PER HOUR LBS. WEIGHT	PER I.H.P	DURATION OF TEST.
6:45:00							
7:06:30	21.30	5	87	86			
7:27:50	21.20	10	206	82			
7:41:16	13.26	15	326	80			
7:55:32	14.16	20	439	78			
8:10:25	14.53	25	566	77			
8:26:10	15.45	30	672	76			
					398.5	155.3	1 Hr. 41 M. 10 S.
8:45:00				70			
9:03:17	18.17	10	206	70			
9:28:00	24.43	20	439	70			
9:51:52	23.51	30	673	70			
10:15:38	23.46	40	906	70			
10:34:50	19.12	FULL	1121	70			
					612.4	161.2	1 Hr. 49 M. 50 S.
10:45:00				68			
11:01:32	16.32	10	206	68			
11:19:40	19.06	20	439	68			
11:37:47	18.07	30	673	68			
11:55:12	17.25	40	906	68			
12:19:05	23.53	FULL	1121	68			
					714.9	102.1	1 Hr. 34 M. 05 S.

TABLE 14

U.S.S. CHESTER
OCTOBER 6-7· 1908

TEST OF FORWARD ENGINE ROOM FIRE AND BILGE PUMP.

DIAM. OF STEAM CYL. 10"
WATER PISTON 9"
STROKE 1⅞
12"

	TIME	STEAM PRESSURE GAGE. BOILER	STEAM CHEST.	VACUUM. AUX. CONDENSER.	BACK PRESSURE IN AUX EXH LINE (GAGE).	DISCHARGE PRESSURE.	REVOLUTIONS OR DOUBLE STROKES	MEAN PRESSURE CYLINDER	I.H.P.	CONDENSED EXHAUST STEAM. TIME OF OBSERVATION	INTERVAL	INCHES IN TANK.	WEIGHT IN TANK / LBS.	TEMPERATURE.	PER HR. LBS. WEIGHT.	PER I.H.P.	DURATION OF TEST.
TEST NO 1	AM 10.55	225	120	21	6.	40				10:55:00	M. S.			70			
	11.05	205	115	20	6	40	542	47.1	120	10:59:20	4.20	10	219	70			
	11.15	220	120	20	6.	40	55.	533	13.7	11:08:45	9.25	20	457	70			
	11.25	225	122	21	6	40	55.	498	12.8	11:17:40	8.55	30	689	70			
	11.35	225	125	21	6	40	55.7	568	14.8	11:26:43	8.55	40	925	70			
	Average	220	120	21	6	40	54.9	52.	13.3	11:34:12	7.29	FULL	1137	70	1740 L.	130.8	Hr. 39M. 12.5.
TEST NO. 2	6.50	210	95	10	7.	20				6:50:00				72			
	7.00	225	92	10	6.	20	57.6	36.5	9.8	7:04:37	14.37	10	219	72			
	7.10	225	95	10	6.	20	59.6	33.3	9.3	7:21:15	16.38	20	457	72			
	7.20	245	82	10	6.	20	57.3	30.4	8.1	7:39:52	18.37	30	689	72			
	7.30	245	80	10	6	20	56.5	30.5	8.1	7:58:55	19.03	40	925	72			
	7.40	240	80	10	6	20	54.8	29.5	7.6	8:14:32	15.37	FULL	1137	72			
	7.50	235	79	10	6	20	54.5	29.8	7.6								
	8.00	225	80	10	6	20	54.5	30.	7.7								
	8.10	235	80	10	6	20	54.3	31.	7.9								
	Average	232	85	10	6.	20	56.1	31.4	8.3						807.1	97.2.	1 Hr. 24 M. 32 S.
TEST NO. 3	8.25	230	56	10	6	20				8:25:00				72			
	8.35		56			20	40.	22.3	4.2	8:40:25	15.25	5	102	72			
	8.45	240	56	10	6.	20	40.			8:57:00	16.35	10	219	72			
	8.55	250	56	10	6.	20	40.1	23.0	4.3	9:13:45	16.45	15	339	72			
	9.05		57			20	41.7			9:30:12	16.27	20	457	72			
	9.15	230	56	10	5	20	39.2	20.8	3.8								
	9.25	230	56	10	5.5	20	40.8										
	Average	236	56	10	5.7	20	40.3	22.0	4.1						420 6	102 5	1 Hr. 05M. 12.5.
TEST NO. 4	9.40	235	80	5	5.	15				9:40:00				70			
	9.50	245	80	5	6.	15	56.7	25.2	6.7	9:50:00	10.00	5	102	70			
	10.00	235	78	5	5.	15	55.6	24.8	6.5	9:57:41	7.41	10	219	70			
	10.10	245	78	5	6.	15	54.7	26.3	6.7	10:07:11	9.30	15	339	70			
	10.20	255	80	5	6.	15	56.8	23.8	6.3	10:15:45	8.34	20	457	70			
	Average	243	79	5	5.6	15	55.9	25.0	6.6	10:25:36	9.31	25	574	70	755.2	114.8	Hr. 45M. 36.5.

TABLE 15

U.S.S. BIRMINGHAM.
OCTOBER 13-1908.

TEST OF E.R. AUX. CONDENSER AIR & CIRCULATING PUMP.

DIAM. OF STEAM CYL. 6"
WATER PISTON RODS. 10"
STROKE 1½
12"

	TIME	STEAM PRESSURE GAGE. BOILER	STEAM CHEST.	VACUUM. AUX. CONDENSER.	BACK PRESSURE IN AUX. EXH. LINE. (GAGE)	REVOLUTIONS.	MEAN PRESSURE CYLINDER	I.H.P.	CONDENSED EXHAUST STEAM. TIME OF OBSERVATION	INTERVAL	INCHES IN TANK.	WEIGHT IN TANK LBS.	TEMPERATURE.	PER HOUR LBS. WEIGHT.	PER I.H.P.	DURATION OF TEST.
TEST NO 1	PM. 4.50	160		26.	6				4:50:00	M. S.			76			
	5.00	170	70	25.5	6	60			5:17:05	27.05	10	216	76			
	5.10		65	25.5	6	60			5:44:22	27.17	20	450	76			
	5.20	170	70	25.5	6	60			5:59:46	15.24	25	569	78			
	5.30	160	70	25.5	6	60			6:11:26	11.40	30	687	78			
	5.40	170	70	25.5	6	60			6:25:11	13.45	35	801	78			
	5.50		65	25.5	6	60	Not indicated.									
	6.00	170	65	25.5	6	60										
	6.10	170	65	25.5	6	60										
	6.20	170	65	25.5	6	60										
	Average	168	67	25.6	6	60								504.9		1 Hr 35 M 11.5

TABLE 15

U.S.S. SALEM
OCTOBER 12-1908

TEST OF E.R. AUX. CONDENSER AIR & CIRCULATING PUMP

DIAM. OF STEAM CYL. 6"
WATER " 10"
PISTON RODS. 1 7/8"
STROKE 12"

	TIME	STEAM PRESSURE GAGE		VACUUM AUX. CONDENSER.	BACK PRESSURE IN AUX. EXH. LINE (GAGE)	REVOLUTIONS.	MEAN PRESSURE		CONDENSED EXHAUST STEAM.						DURATION OF TEST.
		BOILER	STEAM CHEST.				CYLINDER	I.H.P.	TIME OF OBSERVATION	INTERVAL	INCHES IN TANK.	WEIGHT IN TANK. LBS.	TEMPERATURE.	PER HOUR LBS. WEIGHT.	PER I.H.P.
TEST NO. I.	P.M. 10.27	200		10	6.	45.	25.8	1.9	10.28.00	M. S.			70°		
	1037					45.			10.47.10	19.10	5	87	70		
	10.47	200		10	6.	45.9	17.5	1.3	11.14.25	27.15	10	206	70		
	1057					46.7	22.	1.7	11.39.45	25.20	15	325	70		
	1108	200		10	6	45.			12.07.00	27.15	20	439	70		
	1127					40							70		
	1147	200		10	6.	46.4	22.	1.6							
	1157	200		10	6.	44									
	AVERAGE	200		10	6.	44.8	21.8	1.6						266.1	166.3

1 Hr 39 M 0 S.

TABLE 16

U.S.S. CHESTER.
OCTOBER 8-1908.

TEST OF AUXILIARY CONDENSER AIR AND CIRCULATING PUMP.

DIAM. OF STEAM CYL. 6"
WATER " 10"
PISTON RODS. 1 7/8"
STROKE 12"

	TIME	STEAM PRESSURE GAGE		VACUUM. AUX. CONDENSER.	BACK PRESSURE IN AUX. EXH. LINE. (GAGE)	REVOLUTIONS.	MEAN PRESSURE		CONDENSED EXHAUST STEAM.						DURATION OF TEST.
		BOILER	STEAM CHEST.				CYLINDER	I.H.P.	TIME OF OBSERVATION	INTERVAL.	INCHES IN TANK.	WEIGHT IN TANK. LBS.	TEMPERATURE.	PER HR. LBS. WEIGHT.	PER I.H.P.
TEST NO. I.	P.M. 7.00	170	96	22	5.				7.00.00	M. S.			F 72		
	7.10	185	100	22	5.	60.	46.8	4.7	7.13.08	13.08	5	102	72		
	7.20		100			60.8	50.0	4.9	7.28.42	15.34	10	219	72		
	7.30	175	102	22	5.	61.7			7.44.09	15.27	15	339	72		
	7.40	175	100	22	5.	61.			7.58.45	14.36	20	457	72		
	7.50		100			61.1			8.10.28	11.43	25	574	72		
	8.00	195	98	21	5.	57.9	49.0	4.5	8.22.35	12.07	30	689	72		
	8.10	195	100	21	5.	58.7									
	8.20	225	100	21	5.	58.2									
	AVERAGE	189	100	22.	5.	59.9	49.3	4.7						500.5	106.5

1 Hr. 22 M. 35 S

TABLE 17

U.S.S. BIRMINGHAM
SEPTEMBER 29-1908.

TEST OF AUXILIARY FEED PUMP.

DIAM. OF STEAM CYL. 9"
WATER " 6"
PISTON RODS. 1⅞"
STROKE 12"

	TIME	STEAM PRESSURE GAGE: BOILER	STEAM PRESSURE GAGE: STEAM CHEST	VACUUM AUX. CONDENSER.	BACK PRESSURE IN AUX. EXH. LINE (GAGE)	DISCHARGE PRESSURE	REVOLUTIONS OR DOUBLE STROKES	MEAN PRESSURE: CYLINDER	I.H.P.	CONDENSED EXHAUST STEAM: TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK LBS.	TEMPERATURES	PER HOUR LBS: WEIGHT	PER I.H.P.	DURATION OF TEST.
TEST NO. 1	A.M. 10.20	250	150	30	5.	295				10:20:00	M. S.			86			
	10.30	250	150	30	5.	298	144	104.	4.7	10:37:02	17.20	10	217	84			
	10.40	250	150	30	5.	295	138	1105	4.8	10:57:09	1949	20	459	88			
	10.50	250	155	30	5.	325	132	116.5	4.9	11:19:40	22.31	30	688	90			
	11.00	250	155	30	5.	312	131	113.	4.7	11:45:25	25.45	40	922	84			
	11.10	250	155	30	5.	315	138	114.	5.0	12:05:44	2221	FULL	1134	86			
	11.20	250	165	30	5.	330	150	111.	4.6								
	11.40	250	165	30	5.	327	132	112.5	4.7								
	11.50	250	165	30	5.	310	144	115.5	5.3								
	12.00	250	155	30	5.	310	118	116.	4.3								
	12.10	250	155	30	5.	310	122	116.	4.5								
	AVERAGE	250	156	30	5.	311	133	111.7	4.8						643.5	134.1	1 Hr. 45 M. 44 S.

TABLE 17

U. S. S. SALEM
OCTOBER 10-1908.

TEST OF AUXILIARY FEED PUMP FIRE ROOM

DIAM. OF STEAM CYL. 9"
WATER " 6"
PISTON RODS. 1⅞"
STROKE 12"

	TIME	STEAM PRESSURE GAGE: BOILER	STEAM PRESSURE GAGE: STEAM CHEST	VACUUM AUX. CONDENSER.	BACK PRESSURE IN AUX. EXHAUST LINE (GAGE)	DISCHARGE PRESSURE	REVOLUTIONS OR DOUBLE STROKES	MEAN PRESSURE: CYLINDER	I.H.P.	CONDENSED EXHAUST STEAM: TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK LBS	TEMPERATURES	PER HOUR LBS: WEIGHT	PER I.H.P.	DURATION OF TEST
TEST NO. 1	P.M. 7.15	260	265	10	5.	300	36	1498	20.3	7:15:00	M.S.			88			
	7.25	265	260	10	7	300	35	1473	19.4	7:21:24	6.24	10	206	88			
	7.35	265	260	10	7	300	35	1598	21.1	7:28:25	7.01	20	439	88			
	7.45	255	255	10	6.	300	34	1593	20.4	7:35:49	7.24	30	672	88			
	7.55	258	250	10	6.	300	34	1603	20.5	7:43:08	7.19	40	904	86			
	AVERAGE	261	258	10		300	35	1553	20.3	7:50:10	7.02	FULL	1118	86	1907.5	94.0	0 Hr. 35 M. 10 S.
TEST NO. 2	8.10	270	225	10	8.	300	25	154.	14.5	8:10:00				80			
	8.20	265	225	10	8.	300	25	1533	14.5	8:18:48	8.48	10	206	80			
	8.30	260	225	10	6.	300	25	1503	14.2	8:27:47	8.59	20	439	80			
	8.40	263	225	10	6.	300	25	1493	14.1	8:38:00	10.13	30	672	79			
	8.50	270	225	10	6.	300	25	1558	14.7	8:48:10	10.10	40	905	79			
	9.00	265	225	10	6	300	25	1485	14.0	8:57:40	10.30	FULL	1120	79			
	AVERAGE	266	225	10	6.6	300	25	1522	14.3						1409.8	98.6	0 Hr. 47 M. 40 S.
TEST NO. 3	9.05	265	190	10	6.	300	15	1368	7.7	9:05:00				80			
	9.15	260	190	10	6.	300	15	1398	7.9	9:19:22	14.22	10	206	79			
	9.25	255	190	10	6.	300	15	1348	7.6	9:35:17	15.55	20	439	76			
	9.35	265	190	10	6.	300	15	1420	8.0	9:51:12	15.57	30	673	74			
	9.45	260	190	10	6.	300	15	1353	7.7	10:06:35	15.23	40	906	73			
	9.55	263	190	10	6.	300	15	1363	7.7	10:20:15	13.40	FULL	1121	73			
	10.05		190	10	6	300	15	1383	7.8								
	10.15		190	10	6	300	15	1363	7.7								
	10.25	270	190	10	6.	300	15										
	AVERAGE	263	190	10	6.	300	15	1375	7.8						893.8	114.6	0 Hr. 15 M. 15 S.

TABLE 18

U.S.S. CHESTER
OCTOBER 6 1908.

TEST OF AUXILIARY FEED PUMP.

DIAM. OF STEAM CYL. 9"
WATER " 6"
PISTON RODS. 1⅞"
STROKE 12"

	TIME	STEAM PRESSURES. GAGE. BOILER	STEAM CHEST.	VACUUM. AUX. CONDENSER	BACK PRESSURE IN AUX. EXH. LINE. (GAGE)	DISCHARGE PRESSURE.	REVOLUTIONS OR DOUBLE STROKES	MEAN PRESSURE. CYLINDER.	I.H.P.	CONDENSED EXHAUST STEAM. TIME OF OBSERVATION	INTERVAL	INCHES IN TANK.	WEIGHT IN TANK. LBS.	TEMPERATURE.	PER HR. LBS. WEIGHT	PER I.H.P.	DURATION OF TEST.
TEST NO. 1	P.M. 1.35	220	190	10	6.	300				1:38:00	M. S.			F 73			
	1.45	210	185	10	6.	300	30.1	130.	14.7	1:48:28	10.28	10	219	73			
	1.55	230	190	10	6.	300	31.4	128.5	15.2	1:58:55	10.25	20	457	73			
	2.05	235	185	10	6.	300	29.5	136.	15.1	2:06:15	10.22	30	689	73			
	2.15	225	184	10	6.	300	28.8	135.7	14.7	2:17:43	11.28	40	925	73			
	2.25	220	184	10	6.	300	28.7	127.	13.7	2:27:39	9.56	FULL	1137	73	1295.7	88.14	0 Hr. 52 M. 39 S.
	AVERAGE	223	187	10	6.	300	29.7	131.4	14.7								

TABLE 19

U.S.S. BIRMINGHAM
OCTOBER 12-1908

TEST OF FIRE AND BILGE PUMP ENGINE ON ASH EJECTOR.

DIAM OF STEAM CYL. 12"
WATER " 8"
PISTON RODS. 2"
STROKE 12"

	TIME.	STEAM PRESSURE GAGE BOILER	STEAM CHEST	VACUUM. AUX. CONDENSER	BACK PRESSURE IN AUX. EXH. LINE. (GAGE)	DISCHARGE PRESSURE	REVOLUTIONS OR DOUBLE STROKES	MEAN PRESSURE CYLINDER.	I.H.P.	CONDENSED EXHAUST STEAM. TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK LBS.	TEMPERATURE	PER HOUR LBS. WEIGHT	PER I.H.P	DURATION OF TEST
TEST NO. 1	P.M. 4.20	205	180	26	6	280	36			4:22:00				88			
	4.30	195	180	26	6.	280	36	104.	25.3	4:27:50	5.50	10	216	88			
	4.40	195	180	26	6	280	36	104.8	25.5	4:34:53	7.03	20	449	88			
	4.50	195	180	26	6.	280	36	104.8	25.5	4:41:38	6.45	30	686	88			
	5.00	190	180	26	6.	280	36	107.	26.0	4:48:15	6.37	40	920	88			
	AVERAGE	196	180	26	6	280	36	105.2	25.6	4:54:26	6.11	FULL	1136	88	2101.6	82.1	0 Hr. 32 M. 26 S.

TABLE 19

U.S.S. SALEM.
OCTOBER 10 1908.

TEST OF FIRE AND BILGE PUMP ENGINE ON ASH EJECTOR.

DIAM OF STEAM CYL. 12"
WATER " 8"
PISTON RODS 2"
STROKE 12"

	TIME	STEAM PRESSURE GAGE. BOILER	STEAM CHEST	VACUUM AUX. CONDENSER.	BACK PRESSURE IN AUX. EXH. LINE (GAGE)	DISCHARGE PRESSURE.	REVOLUTIONS OR DOUBLE STROKES	MEAN PRESSURE CYLINDER	MEAN PRESSURE I.H.P.	CONDENSED EXHAUST STEAM. TIME OF OBSERVATION	INTERVAL	INCHES IN TANK.	WEIGHT IN TANK LBS	TEMPERATURE	PER HOUR LBS. WEIGHT	PER HOUR LBS. PER I.H.P.	DURATION OF TEST
TEST NO.1	P.M. 1.45	265	260	10	5.	265	50	120.1	40.6	1:45:00	M. S.			106			
	1.55	252	265	10	6.	265	51	118	40.8	1:48:00	3.00	10	205	106			
	2.05	255	265	10	6.	265	49	118.7	39.3	1:51:22	3.22	20	437	106			
	AVERAGE	257	263	10	5.6	265	50.	118.9	40.2	1:54:41	3.19	30	670	106			
										1:58:02	3.21	40	902	106	4193.6	104.3	0Hr. 15M. 58 S.
										2:00:58	2.56	FULL	1116	106			
TEST NO.2	2.15	255	205	10	6	265	36	108.8	26.5	2:15:00				108			
	2.25	260	200	10	6	265	37	98.8	24.7	2:19:25	4.25	10	205	108			
	2.35	260	200	10	6	265	37	95	23.8	2:24:50	5.25	20	437	108			
	AVERAGE	258	202	10	6.	265	36.6	100.8	25.0	2:29:38	4.48	30	670	108			
										2:34:30	4.52	40	902	108	2742.5	109.7	0Hr. 24M. 25 S.
										2:39:25	4.55	FULL	1116	108			
TEST NO.3	2.50	255	165	10	6	265	22	80.1	11.9	2:50:00				90			
	3.00	255	165	10	6.	265	22	94.	14.	2:57:48	7.48	10	205	90			
	3.10	265	165	10	6	265	22	93.8	14.	3:06:48	9.00	20	438	88			
	3.20	265	168	10	6.	265	23.	92.3	14.4	3:15:08	8.20	30	671	86			
	3.30	260	165	10	6	265	22.	101.3	15.1	3:22:40	7.32	40	904	86	1639.4	117.9	0Hr. 40M. 55 S.
	AVERAGE	260	166	10	6.	265	22.2	92.3	13.9	3:30:55	8.15	FULL	1118	84			

TABLE 20

U.S.S. CHESTER.
OCTOBER 6-1908.

TEST OF FIRE AND BILGE PUMP ON ASH EJECTOR.

DIAM. OF STEAM CYL. 9"
WATER " 6"
PISTON RODS 1 7/8"
STROKE 12"

	TIME	STEAM PRESSURE GAGE. BOILER.	STEAM CHEST	VACUUM AUX. CONDENSER.	BACK PRESSURE IN AUX. EXH. LINE (GAGE).	DISCHARGE PRESSURE.	REVOLUTIONS OR DOUBLE STROKES.	MEAN PRESSURE CYLINDER.	MEAN PRESSURE I.H.P.	CONDENSED EXHAUST STEAM. TIME OF OBSERVATION	INTERVAL	INCHES IN TANK.	WEIGHT IN TANK. LBS.	TEMPERATURE.	PER HR. LBS. WEIGHT	PER HR. LBS. PER I.H.P.	DURATION OF TEST.
TEST NO. 1	P.M. 4.40	220	178		6.	250				4:40:00	M. S.			88			
	4.50	220	178		6.	265	36.5	128.4	31.7	4:45:30	5.30	10	219	88			
	5.00	225	175		6.	265	36.3	129.6	31.8	4:51:20	5.50	20	457	88			
	5.10	220	175		6.	265	36.1	128.3	31.3	4:56:13	4.53	30	688	88			
	5.20		175			265	36.3	126.	30.9	5:01:20	5.07	40	924	88	2674.	85.1	0Hr. 25M. 30S.
	AVERAGE	221	176		6.	262	36.3	128.1	31.4	5:05:30	4.10	FULL	1136	88			

TABLE 21

U S S. BIRMINGHAM
OCTOBER 11 1908

TEST OF FORCED DRAFT BLOWER (DOUBLE CYLINDER SIMPLE) ENGINE NO. 2

DIAMETER OF CYLINDERS 6"
STROKE 6"
DIAMETER OF PISTON RODS 1⅛"

	TIME	STEAM PRESSURES GAGE: BOILER	STEAM CHEST OF BLOWER	VACUUM AUX. CONDENSER	BACK PRESSURE IN AUX. EXH. LINE (GAGE)	REVOLUTIONS	AIR PRESSURE INS OF WATER	MEAN PRESSURES: FORD CYL.	AFTER CYL.	I.H.P.	CONDENSED EXHAUST STEAM: TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK LBS	TEMPERATURE	PER HOUR LBS: WEIGHT	PER I.H.P.	DURATION OF TEST
											M. S.							
TEST NO 1	11 20	170	125	3.	6.	380	0	No indicator cards taken.			11:20:00							1 Hr. 18 M. 30 S.
	11 30	165	125	3.	6.3	380	0				11:25:18	5.18	10	216				
	11 40	168	125	3.	5.8	380	0				11:33:35	8.17	20	450	78			
	11 50	170	125	3.	6.	380	0				11:40:50	7.15	30	687	79			
	12 00	175	125	3.	6.5	380	0				11:50:30	9.40	40	921	79			
	12 10	175	125	3.	6.	380	0				11:57:55	7.25	FULL	1138	78			
	12 20	170	125	3.	5.8	380	0				12:04:55	7.00	10	217	80	1736.7		
	12 30	170	125	3.	6.	380	0				12:12:35	7.40	20	455	82			
	12 40	170	125	3.	6.	380	0				12:21:35	9.00	30	688	82			
	AVERAGE	170	125	3.	6	380	0				12:30:37	9.02	40	922	81			
											12:38:30	7.53	FULL	1134	81			
TEST NO 2	1.00	165	110	3.	6.	360	0	No indicator cards taken.			1:00:00							0 Hr. 43 M. 30 S.
	1.10	155	110	3.	6.	360	0				1:08:08	8.08	5	97				
	1.20	165	110	3.	6.	360	0				1:07:50	4.42	10	216				
	1.30	165	110	3.	5.	360	0				1:12:28	4.38	15	334				
	1.40	165	110	3.	6.	360	0				1:17:45	5.17	20	450				
	1.50	160	110	3.	6.	360	0				1:22:50	5.05	25	569				
	AVERAGE	163	110	3.	6	360	0				1:27:05	4.15	30	687				
											1:30:55	3.50	35	801				
											1:35:05	4.10	40	921	79			
											1:39:45	4.40	45	1040	79	1569.7		
											1:48:30	3.45	FULL	1138	79			
TEST NO 3	2.00	165	95	3.	6.	310	0	No indicator cards taken.			2:00:00							1 Hr. 6 M. 15 S.
	2.10	160	95	3.	6.	310	0				2:05:50	5.50	5	97	76			
	2.20	160	95	3.	6.	310	0				2:14:15	8.25	10	216	76			
	2.30	173	95	3.	6.	310	0				2:21:30	7.15	15	334	76			
	2.40	170	95	3.	6.	310	0				2:28:25	6.55	20	450	76			
	2.50	157	95	3.	6.	310	0				2:35:15	6.50	25	569	76			
	3.00	167	95	3.	6	310	0				2:42:45	7.30	30	687	76			
	3.10	167	95	3.	6.	310	0				2:49:15	6.30	35	801	76			
	AVERAGE	165	95	3.	6.	310	0				2:55:25	6.10	40	921	76			
											3:01:15	5.50	45	1040	76	1030.6		
											3:06:15	5.00	FULL	1138	76			
TEST NO 4	3.20	170	75	3.	6.	260	0	No indicator cards taken.			3:20:00							0 Hr. 45 M. 00 S.
	3.30	170	75	3.	6.	260	0				3:27:00	7.00	5	97				
	3.40	170	75	3.	6.	260	0				3:35:30	8.30	10	216				
	3.50	165	75	3.	5.8	260	0				3:44:30	9.00	15	334				
	4.00	165	75	3.	6.5	260	0				3:55:30	11.00	20	450	74			
	4.10	165	75	3.	6.	260	0				4:05:00	9.30	25	569	75			
	AVERAGE	168	75	3.	6.	260	0									758.7		
TEST NO. 5	4.50	170	125	3.	6.	380	0	No indicator cards taken			4:50:00							0 Hr. 36 M. 50 S.
	5.00	170	125	3.	6	380	0				4:52:40	2.40	5	97				
	5.10	160	125	3.	6	380	0				4:56:20	3.40	10	216				
	5.20	150	125	3.	6.	380	0				5:00:05	3.45	15	334				
	5.30	165	120	3.	6	375	0				5:02:55	3.50	20	450				
	AVERAGE	163	124	3.	6.	375	0				5:07:45	3.50	25	569				
											5:11:30	3.45	30	687				
											5:15:50	4.20	35	801				
											5:20:30	4.40	40	921				
											5:24:20	3.50	45	1040				
											5:28:30	2.30	FULL	1138		1653.8		

Tabulated by AMD
Checked by WWW

NOTES.—During all of the above tests the fireroom (No. 1), was closed (hatches, doors, etc.), airtight, except discharge opening to No. 1 blower. This opening allowed free escape of air, and in consequence, no pressure was recorded in fireroom.

Test No. 5 was made for the purpose of checking results obtained in test No. 1, the latter not being considered entirely reliable.

Revolutions of blower were taken by tachometer.

TABLE 22

U.S.S. SALEM.
OCTOBER 11-1908.

TEST OF FORCED DRAFT BLOWER (DOUBLE CYLINDER SIMPLE) ENGINES NOS. 1 & 2.

DIAMETER OF CYLINDERS. 6"
STROKE 6"
DIAMETER OF PISTON RODS. 1⅜"

	TIME	STEAM PRESSURES. GAGE. BOILER	STEAM CHEST. #1 STBD.	STEAM CHEST. #2 PORT	VACUUM (AUX. CONDENSER)	BACK PRESSURE IN AUX. EXH. LINE. (GAGE)	REVOLUTIONS. #1 STBD.	REVOLUTIONS. #2 PORT.	AIR PRESSURE. INS. OF WATER.	MEAN PRESSURES. FORD CYL.	MEAN PRESSURES. AFTER CYL.	I.H.P. ONE ENGINE.
TEST NO. 1	12.55	195	150	150	10	6.	485	492	4.	Indicator card		
	1.05	170	148	150	10	6.	479	486	4.	data unreliable		
	Average	183	149	150	10	6.	482	490	4			
TEST NO. 2		125	275	104	107	10	6.	395	402	2.5	46.9		31.5
		135	275	105	107	10	6.	397	401	2.5	48.25	Not taken.	32.3
		1.45	275	105	107	10	6.	398	402	2.5	48.75		32.8
	Average		275	105	107	10	6.	397	402	2.5	47.97		32.2
TEST NO. 3		1.55	275	52.	55	10	6.	223	222	1.	17.90		6.6
		2.05	285	51	55	10	6	224	223	1.	17.90	Not taken.	6.7
		2.15	275	50	57	10	6	223	221	1.	17.90		6.6
		2.25	280	50	60	10	6.	225	224	1.	18.00		6.7
		2.35	275	50	58	10	6.	224	226	1.	17.20		6.5
		2.40	275	50	57	10	6	225	223	1.	18.00		6.7
	Average		278	51	57	10	6	224	223	1.	17.81		6.7
TEST NO. 4		3.10	275	40	42	10	6	165	163	5	12.90		3.5
		3.20	275	40	42	10	6	164	163	5	11.00		3.0
		3.30	275	40	43	10	6.	163	160	5	11.50	Not taken.	3.1
		3.40	275	40	43	10	6.	163	162	5	12.00		3.2
		3.50	275	40	44	10	6	165	164	5	11.90		3.3
		4.00	275	40	44	10	6	164	165	5	11.90	Not taken	3.3
		4.10	275	—	—	10	6.	—	—	—	—		—
	Average		275	40	43	10	6.	164	166	5	11.87		3.2
TEST NO. 5		4.15	280	125	125	10	6.	427	418	3.	—		
		4.25	280	125	125	10	6	426	428	3.	—	Not taken	
		4.35	275	125	125	10	6	430	432	3.	49.75		36.9
	Average		278	125	125	10	6	428	426	3.	49.75		35.9
TEST NO. 6.		4.50	280	75	78	10	6.	301	304	1.5	32.50		16.5
		5.00	280	75	78	10	6.	306	307	1.5	33.25	Not taken	16.5
		5.10	275	75	80	10	6.	306	308	1.5	32.50		16.9
		5.20	275	75	77	10	6.	308	302	1.5	32.75		16.5
	Average		278	75	78	10	6	305	305	1.5	32.85		16.6

	CONDENSED EXHAUST STEAM. TIME OR OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK LBS.	TEMPERATURE	WEIGHT PER HOUR	PER I.H.P.	DURATION OF TEST
TEST NO. 1	12.55.00	H. M.			129			
	12.57.03	2.03	10	200	125			
	12.59.32	2.29	20	432	122			
	1.02.10	2.38	30	665	119			
	1.04.45	2.35	40	897	118			
	1.07.06	2.21	FULL	1111	118	3509.0		O.W.-12 M.-0.5
TEST NO. 2	12.50.0				97			
	12.80.5	3.05	10	205	97			
	1.31.42	3.37	20	437	97			
	1.35.30	3.48	30	670	96			
	1.39.07	3.37	40	902	96			
	1.42.34	3.27	FULL	1116	96	3811.6	592	O.W.-17 M.-34.5
TEST NO. 3	1.55.00				96			
	1.59.32	4.32	5	90	96			
	2.05.03	5.31	10	209	82			
	2.10.38	5.35	15	329	80			
	2.16.15	5.37	20	441	78			
	2.21.56	5.41	25	568	78			
	2.27.21	5.25	30	674	78			
	2.32.45	5.24	35	788	78			
	2.38.18	5.33	40	906	78			
	2.43.38	5.20	45	1025	78	1255.7	93.7	O.W.-8 M.-31.6.
	2.48.31	4.53	FULL	1120	78			
TEST NO. 4	3.10.00				74			
	3.16.05	6.05	5	91	74			
	3.28.00	11.55	10	210	72			
	3.37.40	9.40	15	330	72			
	3.46.20	8.40	20	442	72			
	3.54.35	8.15	25	569	72			
	4.03.44	9.09	30	675	70			
						755.7	117.0	O.W.-33 M.-44.5.
TEST NO. 5	4.16.00				110			
	4.18.30	2.30	10	205	110			
	4.21.28	2.58	20	435	110			
	4.24.25	2.57	30	668	112			
	4.27.13	2.58	40	900	112			
	4.29.43	2.40	FULL	1114	113	4872.9	67.9	O.W.-13 M.-43.8.
TEST NO. 6.	4.50.00				96			
	4.55.08	5.08	10	206	96			
	5.00.56	5.48	20	436	96			
	5.06.44	5.48	30	671	95			
	5.12.26	5.42	40	903	96			
	5.18.00	5.44	FULL	1117	96	2391.5	72.0	O.W.-2 M.-0.5.

NOTES.—In all the above tests two blower engines were in operation. Only one (forward) cylinder of No. 2 (port) blower, however, was indicated. It is assumed that the total I.H.P. for each blower engine is double that shown for the cylinder indicated.

The two blower engines were run, as nearly as possible, under similar conditions as to steam-chest pressure and revolutions, and the power developed by each is assumed to be the same. It should be noted that the condensed exhaust steam collected is from both engines.

Fireroom No. 1 was closed airtight. In tests 1, 2, 3 and 4, six furnace doors were thrown open (ash-pan doors closed) to provide air leakage. Tests 5 and 6 were made with all (12) furnace doors open, other conditions being similar to tests 1-4.

Revolutions of blowers were taken by speedometer.

TABLE 23
SHEET 1

U.S.S. CHESTER.
OCTOBER 5-1908

TEST OF FORCED DRAFT BLOWER (DOUBLE CYLINDER SIMPLE) ENGINES NOS. 1 & 2.

DIAMETER OF CYLINDERS 5"
STROKE
DIAMETER OF PISTON RODS 1 1/4"

	TIME	STEAM PRESSURE GAGE: BOILER	STEAM PRESSURE GAGE: STEAM CHEST OF BLOWER.	VACUUM (AUX. CONDENSER) BACK PRESSURE IN AUX. EXH. LINE (GAGE)	AIR PRESSURE INS. OF WATER	REVOLUTIONS.	MEAN PRESSURES: FORD. CYL.	MEAN PRESSURES: AFTER CYL.	I.H.P. ONE ENGINE	CONDENSED EXHAUST STEAM: TIME OF OBSERVATION	INTERVAL	INCHES IN TANK.	WEIGHT IN TANK. LBS.	TEMPERATURE.	WEIGHT PER HOUR LBS.	I.H.P.	DURATION OF TEST
TEST NO 1	AM									H. M. S.							
	11.40	210	105	10	5.5	391	4.	56.80	57.96	215	11:40:00	—	—	—	83			
	11.50	210	105	10	6.5	396	4.	57.18	57.86	219	11:42:20	2.20	5	105	83			
	12.00	220	108	10	6.0	393	4.	—	—	—	11:44:48	2.28	10	222	83			
	AVERAGE	213	106	10	6.0	393	4.	56.99	57.91	217	11:47:45	2.57	15	341	83			
											11:50:33	2.48	20	459	83			
											11:53:06	2.33	25	575	83			
											11:55.47	2.41	30	690	83			
											11:58:25	2.28	35	809	83			
											12:01:03	2.38	40	925	83	2604.8	600	0 Hr. 26 M. 10 S.
											12:03:41	2.38	45	1046	83			
											12:06:10	2.31	FULL	1136	83			
TEST NO 2	PM																	
	1.15	210	70	10	4.5	321	2.5	36.16	41.47	120	1.15:00	—	—	—	76			
	1.25	200	70	10	5.5	314	2.5	36.00	37.55	111	1.19:04	4.04	5	105	76			
	1.35	215	70	10	6.0	310	2.5	37.89	38.71	114	1.24:20	5.16	10	222	76			
	1.45	215	70	10	5.5	308	2.5	38.81	40.38	117	1.29:03	4.43	15	341	77			
	1.55	195	70	5	5.5	310	2.5	41.58	41.92	124	1.33:21	4.18	20	459	77			
	AVERAGE	207	70	9	5.4	313	2.5	38.09	40.00	117	1.37:43	4.22	25	575	77			
											1.41:58	4.15	30	690	78			
											1.46:10	4.12	35	809	78			
											1.50:25	4.15	40	925	78			
											1.54.53	4.28	45	1046	78	1631.9	697	0 Hr. 41 M. 46 S.
											1.56:46	1.53	FULL	1136	78			
TEST NO 3	2.10	200	42	10	4.0	188	1.	18.59	18.82	34	2.10.00	—	—	—	76			
	2.20	200	42	10	5.5	188	1.	16.70	16.08	30	2.20.03	10.03	5	105	76			
	2.30	200	42	10	6.0	191	1.	18.59	18.82	34	2.32.22	12.19	10	222	76			
	2.40	200	42	10	5.0	193	1.	18.97	17.63	34	2.42.23	10.01	15	341	76			
	2.50	205	42	10	5.5	192	1.	18.31	17.89	33	2.51.45	9.22	20	459	76			
	3.00	210	42	10	6.0	191	1.	18.24	18.03	33	3.00.16	11.31	25	575	76			
	3.10	205	42	10	5.5	190	1.	18.18	17.76	33	3.18.03	11.47	30	690	76			
	AVERAGE	203	42	10	5.4	190	1.	18.23	17.87	33						696.4	96.4	1 Hr. 5 M. 08 S.
TEST NO 4	4.10	160	52.5	10	5.5	210	1	24.56	23.74	49	4.10.00	—	—	—	90			
	4.20	175	52	10	6.0	208	1	24.00	22.62	47	4.15.03	8.03	5	103	90			
	4.30	210	52	10	6.0	204	1	23.75	23.68	46	4.22.34	7.31	10	220	90			
	4.40	205	52	10	5.0	204	1.	23.28	21.77	44	4.29.27	6.53	15	339	89			
	4.50	205	52	10	6.0	205	1.	24.97	22.95	47	4.37.00	7.33	20	457	88			
	5.00	200	52	10	5.5	204	1.	25.15	23.34	47	4.44.42	7.42	25	573	87			
	5.10	200	52	10	5.0	206	1.	25.96	23.61	49	4.51.48	7.06	30	688	87			
	5.20	215	52	10	5.0	206	1.	23.70	22.56	47	4.59.16	7.28	35	807	86			
	AVERAGE	196	52.1	10	5.5	206	1.	24.66	23.03	47	5.06.09	6.53	40	923	85			
											5.12.48	6.39	45	1044	84			
											5.18.03	5.15	FULL	1134	83			
																1001.6	106.4	1 Hr. 0 M. 03 S.
TEST NO 5	6.00	203	80	10	6.0	320	2.5	50.07	48.42	153	6.00.00	—	—	—	80			
	6.10	215	90	10	5.5	325	2.5	52.41	48.51	157	6.02.55	2.55	5	105	80			
	6.20	240	90	10	6.0	334	2.5	51.45	51.35	165	6.06:19	3.24	10	222	80			
	6.30	240	88	10	6.0	327	2.5	51.00	51.15	160	6.09.38	3.19	15	341	80			
	AVERAGE	224	87.	10	5.9	327	2.5	51.23	50.11	159	6.13.14	3.36	20	459	80			
											6.16.32	3.18	25	575	80			
											6.19.40	3.08	30	690	80			
											6.22.52	3.12	35	809	80			
											6.26.12	3.20	40	925	80			
											6.29.31	3.19	45	1046	80	2120.2	66.7	0 Hr. 32 M. 09 S.
											6.32.09	2.38	FULL	1136	80			

TABLE 23
SHEET 2

U.S.S. CHESTER
OCTOBER 5-1908.

TEST OF FORCED DRAFT BLOWER (DOUBLE CYLINDER SIMPLE) ENGINES NOS. 1 & 2.

DIAMETER OF CYLINDERS 5"
STROKE 5"
DIAMETER OF PISTON RODS 1⅛

	TIME	STEAM PRESSURE GAGE. BOILER	STEAM CHEST OF BLOWER.	VACUUM (ABS. CONDENSER) BACK PRESSURE IN AUX. EXH. LINE (GAGE)	REVOLUTIONS.	AIR PRESSURE INS. OF WATER.	MEAN PRESSURES. FORD CYL.	MEAN PRESSURES. AFTER CYL.	I.H.P. ONE ENGINE	CONDENSED EXHAUST STEAM. TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK LBS	TEMPERATURE	WEIGHT PER HOUR	PER I.H.P.	DURATION OF TEST
TEST NO. 6	P.M. 6.40	230	145	10	5.5	432	4	90.62	—	37.6	6:40:00	M. S			99°			
	6.50	200	145	10	5.5	428	4	81.64	—	33.5	6:41:35	1.35	5	105	95			
	7.00	165	140	10	6.0	411	3.6	81.27	—	32.1	6:43:25	1.50	10	220	93			
	AVERAGE	198	143	10	5.7	424	3.9	84.51	—	34.4	6:45:23	1.58	15	339	93			
											6:47:11	1.48	20	457	93			
											6:49:00	1.49	25	573	93			
											6:50:55	1.55	30	688	93			
											6:53:03	2.08	35	807	93			
											6:55:06	2.03	40	925	93			
											6:57:06	2.00	45	1044	93			
											6:59:15	2.09	FULL	1134	93	3534.5	51.4	0 HR. 19 M. 15 S.
TEST NO. 7	8.20	200	61	10	6.0	222	1.	30.80	30.59	6.5	8:00:00				74°			
	8.30	195	61	10	6.0	216	1.	28.15	29.41	6.0	8:25:28	5.28	5	106	74			
	8.40	200	63	10	6.0	222	1.	31.10	30.79	6.6	8:37:35	6.54	10	223	74			
	8.50	205	65	10	6.0	222	1.	30.30	31.74	6.6	8:50:20	5.59	15	342	74			
	9.00	190	68	10	6.5	227	1.	30.69	31.32	6.8	8:44:03	5.43	20	460	74			
	9.10	200	68	10	6.5	227	1.	31.92	32.79	7.5	8:49:50	5.47	25	576	75			
	9.20	200	68	10	6.0	226	1.	33.24	31.48	7.0	8:56:00	6.10	30	691	75			
	AVERAGE	198	65	10	6.1	223	1.	30.89	31.30	6.7	9:01:05	5.05	35	810	76			
											9:06:36	5.31	40	926	76			
											9:12:14	5.38	45	1047	75			
											9:16:28	4.14	FULL	1137	75	12082	90.2	0 HR. 56 M. 28 S.
TEST NO. 8	9.30	200	130	10	7.0	380	2.5	77.01	76.98	28.1	9:30:00				85°			
	9.40	190	130	10	6.5	382	2.5	—	72.85	26.7	9:31:58	1.58	5	104	85			
	9.50	195	130	10	5.5	384	2.6	—	75.71	27.9	9:33:56	1.57	10	221	85			
	AVERAGE	195	130	10	6.3	382	2.5	77.01	75.17	27.6	9:36:20	2.25	15	340	85			
											9:38:31	2.11	20	458	85			
											9:40:50	2.21	25	574	85			
											9:43:00	2.10	30	689	85			
											9:45:05	2.05	35	808	85			
											9:47:19	2.14	40	924	85			
											9:49:21	2.02	45	1045	85			
											9:51:15	1.54	FULL	1135	85	3204.6	38.1	0 HR. 21 M. 15 S.

TABULATED BY AND CHECKED BY W.W.W.

NOTES.—The blowers for this fireroom (two in number) have a common shaft which is driven by two vertical blower engines connected directly to this shaft. In all the above tests, the forward engine (two cylinders) only was indicated, but both engines were in operation. The power developed was presumably the same for each engine, as by means of steam gauges, the steam-chest pressure was kept precisely the same during each test, for each blower engine.

It is to be noted that the condensed exhaust steam collected is for both engines.

All tests were made with fireroom (No. 1) hatches, etc., closed. In tests 1, 2 and 3, furnace and ash-pan doors, as well as dampers in uptakes were closed; the opening for air leakage in fireroom was therefore the same for each test. Tests 4, 5 and 6 were conducted under similar conditions as to air leakage, and in this respect, differ from tests 1, 2 and 3, only in that dampers in uptakes were open. Tests 6 and 7 were made under conditions similar to tests 4, 5 and 6, except greater air leakage was allowed by opening furnace doors in boilers—five in number.

Revolutions of blowers were taken by speedometer.

TABLE 24

U. S. S. BIRMINGHAM
SEPTEMBER 26-1908

TEST OF FLUSHING PUMP

DIAMETER OF STEAM CYLINDER 10"
DIAMETER OF WATER CYLINDER 10"
DIAMETER OF PISTON ROD 1⅞"
STROKE 12"

	TIME	STEAM PRESSURES GAGE: BOILER	STEAM CHEST	VACUUM AUX. CONDENSER	BACK PRESSURE IN AUX. EXH. LINE (GAGE)	DOUBLE STROKES	MEAN PRESSURE: CYLINDER	I.H.P.	CONDENSED EXHAUST STEAM: TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK LBS.	TEMPERATURE	PER HOUR LBS.: WEIGHT	PER I.H.P.	DURATION OF TEST
TEST NO 1	PM. 6.20	160	46	29	5.5	315	28.75	3.9		H. M. S.						
	6.50	155	46	26	4.3	300	26.00	2.9								
	7.20	150	47	29	4.0	350	30.00	4.0	7:41:30	1.21.30	FULL	1136	80			
	7.50	150	48	29	5.3	320	23.75	3.3								
	8.20	150	45	29	5.0	300	27.00	3.5								
	8.50	135	50	29	5.0	285	28.75	3.5	9:15.00	1.33.30	FULL	1134	80			
	9.15	140	46	28	4.5	280	28.00	3.4								
	AVERAGE	148	47	28	4.8	30.7	27.46	3.5						7790	222.57	2 Hr.-55 M.-0 S.

Discharge pressure of pump = 25 LBS.
Average stroke of pump during test = 11 ins.

TABLE 24

U. S. S. CHESTER
OCTOBER 4-1908

TEST OF FLUSHING PUMP.

DIAMETER OF STEAM CYLINDER 10"
DIAMETER OF WATER CYLINDER 9"
DIAMETER OF PISTON ROD 1⅞"
STROKE 12"

	TIME	STEAM PRESSURES GAGE: BOILER	STEAM CHEST	VACUUM AUX. CONDENSER	BACK PRESSURE IN AUX. EXH. LINE (GAGE)	DOUBLE STROKES	MEAN PRESSURE: CYLINDER	I.H.P.	CONDENSED EXHAUST STEAM: TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK LBS.	TEMPERATURE	PER HOUR LBS.: WEIGHT	PER I.H.P.	DURATION OF TEST
TEST NO 4	PM. 7.30	230	75	10	6.5	54	25.13	6.3	7.30.00	H. M. S.			F 75			
	7.45	225	62	10	6.5	54.8	24.00	6.2	7:40.18	10.18	5	102	75			
	8.00	225	62	10	7.	54.9	23.50	6.0	7:52.34	12.16	10	219	75			
	8.15	225	60	10	6.	54.1	23.75	6.0	8.04.35	12.01	15	339	74			
	8.30	220	60	10	6.8	54.2	22.75	5.8	8:16.28	11.53	20	457	74			
	8.45	221	60	10	6.	54.2	24.00	6.1	8.28.07	11.39	25	574	73			
	9.00	225	60	10	6.	54.3	25.00	6.4	8.38.38	10.31	30	689	72			
	9.15	225	60	10	6.	54.9	24.38	6.2	8.48.51	10.13	35	809	72			
	9.22	225	60	10	6.	56.2			8.59.20	10.29	40	925	72			
									9.10.11	10.51	45	1047	72			
	AVERAGE	225	62	10	6.5	54.6	24.06	6.1	9.21.50	11.39	FULL	1137	72	6100	1000	1 Hr.-51 M.-50 S

Discharge pressure of pump = 12 LBS.
Average stroke of pump during test = 11 ins

TABLE 25

U. S. S. SALEM
OCTOBER 12.-1908.

TEST OF FLUSHING PUMP.

DIAM OF STEAM CYLINDER 10"
" " WATER " 10"
" " PISTON ROD 1¾"
STROKE 12"

	TIME	STEAM PRESSURES (GGS.) BOILER	STEAM CHEST	VACUUM AUX. CONDENSER. BACK PRESSURE IN AUX EXH. LINE (GAGE)	DOUBLE STROKES.	MEAN PRESSURE CYLINDER	I.H.P.	CONDENSED EXHAUST STEAM: TIME OF OBSERVATION	INTERVAL	INCHES IN TANK.	WEIGHT IN TANK LBS.	TEMPERATURE	WEIGHT PER HOUR LBS.	WEIGHT PER I.H.P.	DURATION OF TEST
TEST NO 1	PM 2.15	190	75	10 6.		44.00		2.15.00	H. M.			70°			
	2.25	200	75	10 6	556	39.75	10.4	2.23.44	8.44	10	206	70			
	2.35	198	70	10 6	512	38.75	9.3	2.37.00	13.16	20	439	71			
	2.45	190	70	10 6	533	39.40	9.9	2.51.16	14.16	30	673	71			
	2.55	200	70	10 6	544	39.00	10.0	3.04.05	12.49	40	906	71			
	3.05	185	78	10 5	523	35.90	8.8	3.16.47	12.42	FULL	1121	71			
	3.15	205	72	10 6	530	33.25	8.3	Discharge pressure of pump = 30 LBS.					1068.6	114.6	1 Hr. 1M. 47 S.
	Average	195	73	10 5.9	533	36.58	9.5								
TEST NO 2	4.00	160	57	10 6.		36.25		4.01.00				84			
	4.10	160	60	10 6.	350	36.00	5.9	4.15.15	14.15	10	206	82			
	4.20	190	65	10 8	349	33.00	5.4	4.31.55	15.40	20	439	78			
	4.30	200	67	10 7	358	34.90	5.9	4.47.03	15.08	30	672	76			
	4.40	200	67	10 7.	359	35.90	6.1	5.06.38	19.35	40	905	76			
	4.50	200	65	10 6	351	36.00	6.0								
	5.00	200	60	10 7.	349	35.75	5.9	Discharge pressure of pump = 30 LBS.					827.3	140.2	1 Hr. 05 M 38 S.
	Average	187	63	10 6.7	353	35.40	5.9								
TEST NO 3	5.12	205	46	10 7.		29.25		5.12.00				70			
	5.22	200	45	10 6.	220	33.25	3.1	5.26.20	14.20	5	87	70			
	5.32		45	10	210	33.25	3.3	5.42.42	16.22	10	206	68			
	5.42	200	45	10 6.	200	29.25	2.8	5.58.41	16.09	15	325	68			
	5.52	200	45	10 6.	200	34.00	3.2	6.23.30	25.00	20	439	68			
	6.02	200	45	10 6.	190	34.40	3.1								
	Average	201	45	10 6.2	20.4	32.23	* 3.1	Discharge pressure of pump= 30LBS.					368.4*	118.8*	1 Hr-11M-30 S
TEST NO. 4	6.22	200	90	10 6	350	33.25	8.8	6.22.00				70			
	6.32	200	85	10 7.	320	43.25	7.4	6.36.48	14.48	10	206	68			
	6.42	200	85	10 6.	320	50.75	7.6	6.50.55	14.07	20	439	68			
	6.52	200	90	10 7.	340	43.25	6.9	7.04.02	13.07	30	673	68			
	7.02	200	90	10 4	340	46.25	7.4	7.19.55	15.53	40	906	68			
	7.12	200	89	10 7.	330	46.00	7.1	7.32.07	12.12	FULL	1121	68			
	7.22	200	85	10 7.	320	52.75	7.9	Discharge pressure of pump = 50 LBS.					959.3	126.2	1 Hr 10M. 07 S.
	Average	200	88	10 6.3	330	48.79	7.6								
TEST NO. 5	7.40	200	45	10 6	350	20.75	3.4	7.40.00				74			
	7.50	200	45	10 6	340	19.25	3.1	8.04.35	24.35	10	206	68			
	8.00	200	45	10 6	350	22.00	3.6	8.19.15	14.40	20	439	68			
	8.10	200	43	10 6	317	25.00	3.7	8.43.12	23.57	30	673	68			
	8.20	200	44	10 6	362	26.75	4.6	9.00.40	17.28	40	906	68			
	8.30	200	44	10 6	364	22.90	3.9	9.19.52	19.12	FULL	1121	68			
	8.40	200	45	10 6	342	24.75	4.0	Discharge pressure of pump = 20 LBS.					673.5	177.2	1 Hr. 39 M. 52 S.
	8.50	200	44	10 6	318										
	9.00	200	48	10 6	352										
	9.10	205	50	10 6	354	22.50	3.8								
	9.20	200	48	10 6	350										
	Average	200	45	10 6	345	22.99	3.8								

Average stroke of pump during tests = 12 ins.

*The results of this test are considered unreliable.

TABLE 26

TESTS OF EVAPORATING AND DISTILLING PLANTS

TUBE HEATING SURFACE = 223 SQ FT, EACH EVAPORATOR

U.S.S. BIRMINGHAM. OCTOBER 13-1908.

TIME	STEAM PRESSURE (GAGE) BOILER	IN COILS OF EVAPORATORS.	ON SHELL OF EVAPORATORS.	BACK PRESS IN AUX. EXH. LINE.	VACUUM. AUX. CONDENSER.	DOUBLE STROKE PER MIN. DISTILLER CIRCULATING PUMP.	EVAPORATOR FEED PUMP	FRESH WATER PUMP	CONDENSED EXHAUST STEAM. TIME OF OBSERVATION H. M. S.	INTERVAL. H. M. S.	NUMBER OF FULL TANKS.	TOTAL WEIGHT IN POUNDS.	TEMPERATURE	POUNDS OF WATER PER HOUR DISTILLED PER SQ FT OF TUBE HEATING SURFACE.	LBS. OF WATER DISTILLED PER LB. OF STEAM USED BY EVAPORATING PLANT.	POUNDS OF WATER DISTILLED
12.00	215	50	5	6	215	37	5		12.00.00							
12.30	160	50	5	6	20.	36	10		12.21.54	21.54	1	1135	72			
1.00	175	50	5	6	21.	36.	9		12.41.48	19.54	2	2573	80			
1.30	175	50	5	6	21.	36	11		1.02.15	20.27	3	3611	79			
2.00	180	50	5	6	21.	36.	6		1.22.33	20.18	4	4545	81			
2.30	175	50	5	6	21.	36.	7		1.44.18	21.45	5	5495	80			
3.00	175	50	5	6	21.	36	6		2.04.53	20.35	6	6417	84			
3.30	175	50	5	6	21.	36.	10		2.25.00	20.07	7	7355	82			
4.00	180	50	5	6	21.	36.	8		2.52.55	27.55	8	9089	80			
AVERAGE	179	50	5	6	21.	36.	8	USED INTERMITTENTLY	3.12.11	19.16	9	10021	78			
									3.29.49	17.38	10	11361	79			
									3.54.56	25.07	11	12459	80			
									4.20.00	25.04	12	12763	80	$\frac{10433}{2\cdot223\cdot4} = 5.85$	$\frac{10433}{12763} = .82$	10435.

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8.00	260	45	10.	6	10	37	16		8.00.00				73			
8.30	260	40	10.5	6	10	34	16		8.11.85	11.85	1	1137	72			
9.00	275	42	10.	6	10	35	23		8.22.29	10.48	2	2254	72			
9.30	270	35	7.	6	10	35	14		8.36.52	14.23	3	3375	66			
10.00	260	40	9.	6	10	35	14		8.50.06	13.14	4	4509	64			
10.30	265	40	9.	6	10	35	10		9.09.13	19.07	5	5631	64			
11.00	270	40	10.	6	10	35	10		9.28.45	19.32	6	6765	64			
11.30	265	40	10.	6	10	35	10		9.52.41	23.56	7	7897	64			
12.00	265	40	10.	6	10	35	10		10.14.26	21.45	8	9021	64			
AVERAGE	266	40	9.5	6	10	36	14	USED INTERMITTENTLY	10.36.42	22.16	9	10143	62			
									10.57.52	21.10	10	11277	61			
									11.18.05	20.13	11	12392	60			
									11.37.45	19.40	12	13532	60			
									11.52.04	14.19	13	14555	60			
									12.14.00	21.56	14	15709	62			
									12.30.00	16.00	14½	16560	60	$\frac{11921}{2\cdot223\cdot4} = 6.68$	$\frac{11921}{16560} = .72$	11921.

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10.30	225	40	11	6.	10	58	13		10.30.00							
11.00	220	40	11	6	10	58	13		10.45.30	15.30	1	1137	67			
11.30	220	40	11	6	10	57	12		10.57.10	11.40	2	2271	67			
12.00	225	40	11	6	10	57	12		11.10.26	13.15	3	3414	68			
12.30	225	40	11	6	10	57	12		11.24.10	13.32	4	4556	66			
1.00	220	40	11	7	10	57	10		11.37.50	13.40	5	5692	66			
1.30	220	40	11	6.5	10	57	12		11.51.55	14.05	6	6832	66			
2.00	210	40	11	6.	10	58	12		12.05.50	13.55	7	7969	66			
2.30	225	40	11	6.	10	58	12	USED INTERMITTENTLY	12.21.40	15.50	8	9109	68			
AVERAGE	221	40	11	6.2	10	58	12		12.39.40	18.00	9	10142	68			
									12.55.10	15.30	10	11286	69			
									1.08.35	13.25	11	12515	68			
									1.19.30	10.55	12	13663	70			
									1.31.40	12.10	13	14800	70			
									1.44.20	12.40	14	15940	70			
									1.56.40	12.20	15	17071	70			
									2.09.30	12.50	16	18217	68			
									2.23.50	14.20	17	19364	69			
									2.39.00	15.10	17½	19957	69	$\frac{14751}{2\cdot223\cdot4} = 8.27$	$\frac{14751}{19957} = .74$	14751

NOTE—The tests above tabulated (each of four hours' duration), were undertaken with a view of establishing a ratio between the weight of boiler steam used and distilled water produced with the plant operating under service conditions.

Of the four evaporators fitted on each vessel, two were used in all tests, and as the tests progressed, each evaporator was blown down and refilled once in four hours in order to simulate conditions of normal use. All evaporator coils were clean. The condensed exhaust water collected included that from evaporator coils, and as well, all steam used by various pumps (distiller circulating, evaporator feed and fresh water—the latter being used as required for pumping distilled water from filter to ships' tanks) installed as a necessary part of the distilling and evaporating plant. During the tests all distilled water was pumped to the ships' tanks, and there measured in gallons, as follows:

BIRMINGHAM, 1,252. *SALEM*, 1,430.55. *CHESTER*, 1,770.

TABLE 27

U. S. S. BIRMINGHAM
SEPTEMBER 27-1908.

TEST OF DYNAMO (COMPOUND) ENGINE NO. 3.

DIAMETER OF H.P. CYLINDER 7½"
DIAMETER OF L.P. CYLINDER 12"
DIAMETER OF PISTON ROD 1⅞"
STROKE 8"

	TIME	STEAM PRESSURE GAGE. BOILER	DYNAMO ROOM	VACUUM AUX. CONDENSER.	REVOLUTIONS	ELECTRICAL LOAD. AMPERES	VOLTS	MEAN PRESSURE. H.P	L.P	I.H.P	CONDENSED EXHAUST STEAM. TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK LBS.	TEMPERATURE	PER HOUR LBS. WEIGHT	PER K.W	PER I.H.P.	DURATION OF TEST.
TEST NO 1	AM 10.00	170	110	14*	400	75	125				10:00:00	M. S.			83				
	10.10	170	110	16	400	75	125				10:12:28	12.28	10	217	80				
	10.20	175	110	16	400	75	125				10:27:51	15.23	20	453	79				
	10.30	175	105	15	400	75	125				10:41:35	13.44	30	688	78				
	10.40	175	108	15	400	75	125				10:56:14	14.39	40	922	78				
	10.50	175	108	15	400	75	125				11:08:45	12.51	FULL	1134	78				
	11.00	173	108	15	400	75	125												
	11.10	185	108	15	400	75	125									989.6			
	Average	175	108	15*	400	75	125			720.4							105.5	48.5	1 Hr - 8 M - 45 S.
TEST NO 2	11.20	175	107	15	400	120	125				11:20:00				79				
	11.30	175	105	15	400	120	125				11:31:38	11.38	10	217	79				
	11.40	178	108	14	400	120	125				11:44:19	12.41	20	453	80				
	11.50	175	108	15	400	120	125				11:58:21	14.02	30	688	75				
	PM 12.00	165	108	15	400	120	125				12:11:28	13.07	40	922	78				
	12.10	180	108	14*	400	120	125				12:22:40	11.12	FULL	1134	80				
	12.20	175	106	15*	400	120	125												
	Average	175	107	15	400	120	125			729.9						1085.7	72.4	36.3	1 Hr - 2 M - 40 S.
TEST NO 3	12.50	175	108	15	400	170	125				12:50:00				88				
	1.00	180	108	14*	400	170	125				12:59:40	9.40	10	217	84				
	1.10	176	108	15	400	170	125				1:10:31	10.51	20	453	83				
	1.20	175	108	15	400	170	125				1:20:37	10.06	30	688	83				
	1.30	176	108	15	400	170	125				1:32:17	11.40	40	922	83				
	1.40	177	108	15	400	170	125				1:42:04	9.47	FULL	1134	83				
	Average	177	108	15	400	170	125			932						1306.7	61.5	33.3	0 Hr - 52 M - 45 S.
TEST NO 4	2.00	170	108	15	400	210	125				2:00:00				87				
	2.10	180	108	15	400	210	125				2:08:28	8.28	10	217	87				
	2.20	178	108	15	400	210	125				2:17:15	8.47	20	453	87				
	2.30	178	108	15	400	210	125				2:26:54	9.39	30	688	86				
	2.40	176	108	15	400	210	125				2:35:56	9.02	40	922	86				
	Average	176	108	15	400	210	125			466	2:43:40	7.44	FULL	1134	86	1558.1	59.4	33.4	0 Hr - 43 M - 40 S.
TEST NO 5	3.15	172	108	15	395	260	125				3:15:00				91				
	3.25	176	108	15	395	260	125				3:31:26	6.26	10	216	93				
	3.35	175	108	15	395	255	125				3:38:29	7.03	20	451	96				
	3.45	176	110	15	395	250	125				3:35:32	7.03	30	686	94				
	Average	175	109	15	395	256	125			557	3:42:46	7.14	40	919	94				
											3:49:00	6.14	FULL	1130	93	1934.1	62.3	35.8	0 Hr - 34 M - 05 S.
TEST NO 6.*	4.10	175	108	15	400	75	125				4:10:00				94				
	4.20	177	108	15	400	78	125				4:18:02	8.02	10	217	90				
	4.30	177	108	15	400	78	125				4:27:13	9.11	20	453	89				
	4.40	175	108	15	400	78	125				4:36:48	9.35	30	688	88				
	4.50	172	108	15	400	75	125				4:44:50	8.02	40	922	87				
	Average	175	108	15	400	77	125			208	4:52:41	7.51	FULL	1134	87	1594.0	63.0	38.3	0 Hr - 42 M - 41 S.
TEST NO 7	5.10		110	15	400	70	125				5:10:00				83				
	5.20	178	108	15	400	70	125				5:24:58	14.58	10	217	81				
	5.30	174	108	15	400	75	125				5:42:13	17.15	20	455	80				
	5.40	175	108	15	400	75	125				5:59:45	17.32	30	688	80				
	5.50	175	108	15	400	78	125				6:17:48	18.03	40	922	79				
	6.00	175	108	15	400	78	125				6:33:03	15.15	FULL	1134	80				
	6.10	175	108	15	400	79	125												
	6.20	177	108	15	400	80	125												
	6.30	178	108	15	400	80	125									819.3			
	Average	176	108	15	400	76	125			706						862		39.8	1 Hr - 2.5 M - 3.5

Indicator cards taken during these tests were unsatisfactory by using the same ratio of K. W. to I.H.P. as was found in corresponding tests on the Chester. The I.H.P. has been calculated

*Nos. 2 and 3 dynamo engines in operation each developing the ampèrage and I.H.P above recorded.
*No. 2 dynamo engine in operation only.

U.S.S. SALEM.
OCTOBER, 8-1908

TABLE 28
TEST OF DYNAMO ENGINE NO. 2.
DIAMETER OF H.P. CYLINDER 7½"
DIAMETER OF L.P. CYLINDER. 12"
DIAMETER OF PISTON ROD 1⅜"
STROKE 8"

	TIME	STEAM PRESSURE. BASE. BOILER.	DYNAMO ROOM.	VACUUM. AUX. CONDENSER.	REVOLUTIONS.	ELECTRICAL LOAD. AMPERES	VOLTS	MEAN PRESSURE. H.P. CYL.	L.P. CYL.	I.H.P.	CONDENSED EXHAUST STEAM. TIME OF OBSERVATION	INTERVAL. M. S.	INCHES IN TANK.	WEIGHT IN TANK LBS.	TEMPERATURE.	PER HOUR LBS. WEIGHT	PER K.W.	PER I.H.P.	DURATION OF TEST
TEST NO. 1	A.M. 1.35	239	110	12	400	75	125				A.M.				F				
	11.45	241	110	20	400	75	125				11.35.00				78				
	11.55	240	111	10	400	75	125				11.43.59	8.59	5	86	78				
	P.M. 12.05	240	110	10	400	75	125				11.55.26	9.27	10	206	78				
	12.15	240	110	10	400	75	125				P.M. 12.04.12	10.46	15	325	78				
	12.25	240	108	10	400	75	125				12.16.00	11.48	20	439	78				
	12.35	240	110	10	400	75	125				12.27.10	11.10	25	566	76				
	12.45	240	109	10	400	75	125				12.38.13	11.03	30	672	73				
	12.55	240	110	10	400	75	125				12.48.12	9.59	35	787	73				
	1.05	240	112	10	400	75	125				1.00.10	12.08	40	908	73				
	1.15	240	110	10	400	75	125				1.12.20	12.00	45	1023	73				
	Average	240	110	11	400	75	125			20.4	1.21.50	9.30	FULL	1120	75	629.0	67.1	30.8	1 Hr - 46 M - 50 S.
TEST NO. 2	1.35	240	110	10	400	120	125				1.35.00				69				
	1.45	240	111	10	400	120	125				1.40.45	5.45	5	86	69				
	1.55	240	109	10	400	120	125				1.50.03	9.18	10	206	69				
	2.05	240	110	10	400	120	125				1.59.40	9.37	15	325	70				
	2.15	240	109	10	400	120	125				2.08.46	9.06	20	439	70				
	2.25	240	110	10	400	120	125				2.17.48	9.02	25	566	70				
	2.35	240	110	10	400	120	125				2.27.05	9.17	30	672	70				
	2.45	240	112	10	400	120	125				2.38.35	8.50	35	787	70				
	2.55	240	110	10	400	120	125				2.44.40	9.05	40	905	70				
	Average	240	110	10	400	120	125			29.9	2.53.08	8.28	45	1023	70				
											3.01.00	7.52	FULL	1120	70	781.4	52.1	26.1	1 Hr - 26 M - 0 S.
TEST NO. 3	3.20	240	110	10	400	170	125				3.10.00				77				
	3.30	245	110	10	400	170	125				3.24.28	4.18	5	86	77				
	3.40	245	111	10	400	170	125				3.31.10	6.42	10	206	76				
	3.50	245	111	10	400	170	125				3.35.50	4.40	15	325	76				
	4.00	240	112	10	400	170	125				3.41.46	5.56	20	439	76				
	4.10	240	112	10	400	170	125				3.49.50	7.04	25	566	76				
	4.20	245	110	10	400	170	125				3.56.47	6.57	30	672	76				
	Average	243	111	10	400	170	125			39.2	4.03.12	6.25	35	787	76				
											4.09.39	6.27	40	905	76				
											4.16.15	6.40	45	1023	76				
											4.22.08	5.53	FULL	1120	76	1081.5	50.9	27.6	1 Hr - 2 M - 8 S.
TEST NO. 4	4.55	245	112	10	400	210	125				4.55.00				76				
	5.05	245	112	10	400	210	125				4.59.15	4.15	5	86	76				
	5.15	245	112	10	400	210	125				5.04.27	5.12	10	206	76				
	5.25	240	112	10	400	210	125				5.10.01	5.35	15	325	76				
	5.35	240	112	10	400	210	125				5.16.30	6.29	20	439	76				
	5.45	240	112	10	400	210	125				5.19.32	3.54	25	566	76				
	Average	243	112	10	400	210	125			46.6	5.23.38	3.06	30	672	77				
											5.29.09	5.31	35	787	77				
											5.34.47	5.33	40	905	77				
											5.40.00	5.18	45	1023	77				
											5.46.18	6.18	FULL	1120	77	1310.0	43.9	28.1	0 Hr - 51 M - 18.5
TEST NO. 5	6.00	240	112	10	400	256	125				6.00.00				77				
	6.10	240	112	10	400	256	125				6.03.10	3.18	5	86	77				
	6.20	245	113	10	400	256	125				6.08.18	5.00	10	206	77				
	6.30	250	112	10	400	256	125				6.13.15	4.53	15	325	77				
	6.40	245	113	10	400	256	125				6.18.09	4.48	20	439	77				
											6.22.54	4.45	25	566	77				
											6.27.40	4.46	30	672	77				
											6.32.33	4.53	35	787	77				
											6.37.13	4.51	40	905	77				
											6.42.18	5.05	45	1023	77				
	Average	244	112	10	400	256	125			55.1	6.46.18	4.10	FULL	1120	77	1446.2	45.2	26.0	0 Hr - 46 M - 28.5

Indicator cards were taken during these tests but were unsatisfactory. The I.H.P. has been calculated by using the mean results of K.W. to I.H.P. as was found in corresponding tests on the chester.

TABLE 29
SHEET 1

U.S.S. CHESTER.
OCTOBER 4 1906.

TEST OF DYNAMO ENGINE NO. 2.

DIAM OF H.P CYL. 7½"
" " L.P. " 12"
" PISTON ROD 1⅝"
STROKE 8"

	TIME	STEAM PRESSURES. GAGE. BOILER	STEAM CHEST	VACUUM AUX. CONDENSER	REVOLUTIONS	ELECTRICAL LOAD. AMPERES.	VOLTS.	MEAN PRESSURES. H.P	L.P	I.H.P	CONDENSED EXHAUST STEAM. TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK. LBS.	TEMPERATURE	PER HOUR LBS. WEIGHT	PER K.W	PER I.H.P	DURATION OF TEST.
TEST NO 1	9.50	210	100		382	218	125	37.50	4.13	49.1	9.50.00				78				
	10.00	200	100		382	220	125	38.25	4.00	49.4	9.59.12	9.12	10	219	78				
	10.10	205	100		382	220	125	37.75	3.88	49.0	10.10.30	11.18	20	457	77				
	10.20	205	100		382	219	125	37.25	4.13	49.0	10.21.42	11.12	30	686	76				
	10.30	203	100		382	220	125	37.75	2.93	47.3	10.32.38	10.56	40	925	74				
	10.40	200	100		382	220	125	36.75	3.25	47.4	10.43.05	11.27	FULL	1137	73				
	AVERAGE	204	100		382	220	125	37.54	3.72	48.5						1285.5	467	26.5	0 Hr. 53 M. 05 S.
TEST NO 2	11.00	205	100		381	255	124	43.25	4.63	53.7	11.00.00				74				
	11.10	200	100		381	250	120	43.00	4.38	53.2	11.04.40	9.40	10	219	74				
	11.20	200	100		381	255	124	45.25	5.13	55.9	11.18.28	8.48	20	457	73				
	11.30	200	100		381	245	125	46.25	3.75	54.4	11.27.52	9.24	30	689	72				
	11.40	200	100		381	250	123	44.25	4.00	53.4	11.37.05	9.11	40	925	72				
	11.50	200	100		381	250	123	43.75	3.38	52.0	11.44.43	7.40	FULL	1137	72				
	AVERAGE	201	100		381	251	123	44.29	4.21	53.8						1525.6	494	28.4	0 Hr. 44 M. 43 S.
TEST NO 3	12.20	205	102		386	170	125	25.88	10.63	35.8	12.20.00				70				
	12.30	210	100		386	175	125	33.50	10.50	40.7	12.30.01	10.01	10	219	70				
	12.40	205	100		386	175	125	34.88	10.88	42.3	12.42.20	12.19	20	457	70				
	12.50	205	100		386	185	124	32.75	11.00	41.1	12.54.20	12.00	30	689	71				
	1.00	200	100		386	185	124	35.00	11.00	42.6	1.06.13	11.53	40	925	71				
	1.10	195	100		386	190	124	33.75	11.00	41.8	1.16.52	10.39	FULL	1137	71				
	1.20	195	100		386	175	124	30.75	10.75	39.5									
	AVERAGE	202	100		386	179	124	32.36	10.82	40.5						1207.7	544	29.8	0 Hr. 56 M. 52 S.
TEST NO 4	1.50	200	95		388	75	125	16.00	6.40	21.8	1.50.00				71				
	2.00	200	95		388	75	125	16.00	5.50	20.4	2.16.58	26.58	10	219	71				
	2.10		95		388	75	125	15.90	5.33	20.7	2.49.22	32.24	20	457	71				
	2.20	205	96		388	75	125	15.50	5.15	19.8	3.17.28	28.06	30	689	71				
	2.30	200	96		388	75	125	15.25	5.00	19.1	3.43.45	26.17	40	925	72				
	2.40		96		388	75	125	16.25	5.25	20.3	4.10.30	26.45	FULL	1137	72				
	2.50	200	96		388	75	125	17.00	5.18	20.6									
	3.00	195	97		388	75	125	16.30	5.55	20.7									
	3.10		96		388	75	125	15.00	5.30	19.4									
	3.20	200	97		388	75	125	16.75	5.25	20.7									
	3.30	190	97		388	75	125	17.00	5.22	20.6									
	3.40	195	97		388	75	125												
	3.50		97		388	75	125												
	4.00	210	100		388	75	125	16.75	5.00	20.0									
	4.10	210	97		388	75	125	16.88	5.25	20.6									
	AVERAGE	200	96		388	75	125	16.18	5.34	20.4						*485.6	*51.8	*23.8	2 Hr. 20 M. 30 S.
TEST NO 5	4.30	225	100		386	120	125	24.2	7.5	29.3	4.30.00				73				
	4.40	225	100		386	120	125	24.8	7.8	30.2	4.54.27	24.27	10	219	73				
	4.50	230	100		386	120	125	25.0	7.7	30.2	5.09.36	16.09	20	457	73				
	5.00	240	100		386	120	125	25.0	7.9	30.5	5.25.00	15.24	30	689	73				
	5.10	235	100		386	120	125	24.1	7.8	29.7	5.42.06	17.06	40	925	73				
	5.20	235	100		386	122	125	23.5	7.7	29.2	6.20.55	36.59	FULL	1137	73				
	5.30	235	100		386	121	125	24.9	8.0	30.6									
	5.40	235	100		386	120	125	25.1	7.9	30.6									
	5.50		100		386	122	126	24.5	7.9	30.2									
	6.00	225	100		386	122	125	24.6	8.0	30.6									
	6.10		100		386	120	125	23.8	8.3	30.4									
	6.20		100		386	122	125	23.9	8.0	30.0									
	AVERAGE	231	100		386	121	125	24.4	7.9	30.1						*616.9	*40.8	*20.5	1 Hr. 50 M. 35 S.

*Unreliable.

TABLE 29
SHEET 2
TEST OF DYNAMO ENGINE NO. 2

U.S.S. CHESTER
OCTOBER 7-8-1908.

	TIME	STEAM PRESSURES GAGE: BOILER	STEAM CHEST	VACUUM AUX. CONDENSER	REVOLUTIONS	ELECTRICAL LOAD: AMPERES	VOLTS	MEAN PRESSURES: H.P.	L.P.	I.H.P.	CONDENSED EXHAUST STEAM: TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK LBS.	TEMPERATURE	PER HOUR LBS.: WEIGHT	PER K.W.	PER I.H.P.	DURATION OF TEST
TEST NO 6	PM 350	220	100	10	387	120	125				3.5000	H. S.			77				
	400	220	102	10	387	120	125				4.0503	15.03	10	219	77				
	410	230	101	10	387	120	125				4.22.10	17.07	20	457	74				
	420	225	100	10	387	120	125				4.37.55	15.45	30	689	69				
	430	225	100	10	387	120	125				4.53.42	15.47	40	925	68				
	440	215	101	10	387	120	125				5.07.55	14.13	FULL	1137	68				
	450	210	101	10	387	120	125												
	500	220	101	10	387	120	125												
	510	215	101	10	387	120	125									875.5	58.3	29.1	1 Hr. 17M 55S.
	AVERAGE	220	101	10	387	120	125	taken	Tests	used.									
TEST NO 7	250	220	100	10	393	75	125				2.5000	H. S.			70				
	300	220	101	10	393	75	125				3.1050	20.50	10	219	70				
	310	220	101	10	393	75	125				3.31.55	21.05	20	457	70				
	320	230	102	10	393	75	125				3.54.04	22.09	30	689	70				
	330	225	102	10	393	75	125	Not	Average	4 and 5	4.15.13	21.09	40	925	70				
	340	225	105	10	393	75	125				4.35.27	18.14	FULL	1137	70				
	350	230	105	10	393	75	125												
	400	220	104	10	393	75	125												
	410	225	105	10	393	75	128												
	420	215	104	10	393	75	125												
	430	225	105	10	393	75	125												
	AVERAGE	223	103	10	393	75	125									659.4	70.3	32.3	1 Hr. 43M 27S.

NOTE.—Tests No. 4 and No. 5 (75 and 120 ampères) unreliable as to steam consumption. An iron chip was discovered, after completion of these tests, under drain valve to measuring tank, which prevented closing the valve properly. This permitted constant leakage from the measuring tank as tests progressed. Because of this, tests No. 6 and No. 7 were made at the same electrical loads, viz: 120 and 75 ampères. No indicator cards were taken on the latter tests, the I.H.P. of tests No. 4 and No. 5 being considered as sufficiently close to determine the power developed.

TABLE 30

U.S.S. BIRMINGHAM
OCTOBER 11-1908

TESTS OF ICE MACHINES, ONE TON "ALLEN" DENSE AIR (DIAM. OF STEAM CYL. ___ PISTON AREA ___ STROKE ___)

	TIME	STEAM PRESSURE GAGE: BOILER	STEAM CHEST STBD.	STEAM CHEST PORT	VACUUM AUX. CONDENSER	BACK PRESSURE IN AUX. EXHAUST LINES	COMPRESSION PRESSURE: STARBOARD	PORT	EXPANSION PRESSURE: STARBOARD	PORT	REVOLUTIONS: STARBOARD	PORT	TEMP COLD SIDE: STARBOARD	PORT	CONDENSED EXHAUST STEAM: TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK LBS.	TEMPERATURE	PER HOUR LBS.: WEIGHT	PER MACHINE	DURATION OF TEST
TEST NO. 1.	AM 900	175	70	75	3	6	210	150	50	50	114	112			9.00.00	H. S.			76			
	910	175	70	77	3	6	210	190	50	50	116	116			9.09.40	9.40	10	216	76			
	920	170	72	75	3	6	210	190	50	50	118	120			9.20.25	10.45	20	450	76			
	930	177	75	77	3	6	210	182	50	48	120	121			9.30.35	10.10	30	687	76			
	940	175	68	77	3	6	205	182	45	48	122	120			9.41.25	10.50	40	921	76			
	950	175	65	78	3	6	208	192	50	46	120	122			9.50.58	9.30	FULL	1138	77			
	AVERAGE	175	70	77	3	6	209	188	45	49	118	118								1341.0	670.5	10 Hr. 50M. 58S.

TABLE 30

U S S SALEM
OCTOBER 12 1908

TESTS OF ICE MACHINES, ONE TON "ALLEN" DENSE AIR
DIAM OF STEAM CYL 8"
PISTON ROD 2 7/8"
STROKE 8"

	TIME	STEAM PRESSURE GAGE. BOILER	STEAM CHEST STBD.	STEAM CHEST PORT	VACUUM CONDENSER BACK PRESSURE IN AUX. EXH. LINE. (GAGE)	COMPRESSION PRESSURE STARBOARD	COMPRESSION PRESSURE PORT	EXPANSION PRESSURE STARBOARD	EXPANSION PRESSURE PORT	REVOLUTIONS STARBOARD	REVOLUTIONS PORT	TEMP. COLD STOR STARBOARD	TEMP. COLD STOR PORT	CONDENSED EXHAUST STEAM. TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK LBS.	TEMPERATURE	WEIGHT PER HR LBS.	WEIGHT PER MACHINE	DURATION OF TEST
TEST NO 1	AM 1045	210	45	55	10 6	125	130	35	35			56	54	104500	M S			86			
	1055	212	45	55	10 6	125	130	35	35	910	989	36	54	105906	1504	10	206	86			
	1105	205	48	50	10 6	125	135	35	34	94	96	40	50	111250	1510	20	689	76			
	1115	200	48	52	10 6	130	130	36	34	95	95	40	50	112522	1532	30	673	77			
	1125	220	48	52	10 6	135	130	36	35	96	96	40	40	114015	1533	40	905	76			
	1135	215	45	50	10 6	140	155	40	36	984	985	40	44	115408	1353	FULL	1120	76			
	1145	220	45	50	10 7	145	155	45	45	995	977	40	44								
	1155	215	45	55	10 6	148	160	45	45	912	988	40	42								
	AVERAGE	212	46	52	10 61	134	141	38	37	946	971	39	47						972.2	486.1	1 Hr 39M 00S

TABLE 31

U S S. CHESTER
OCTOBER 2, 1908

TEST OF ICE MACHINES, ONE TON "ALLEN" DENSE AIR
DIAM OF STEAM CYL 8"
PISTON ROD 1 7/8"
STROKE 8"

	TIME	STEAM PRESSURE GAGE. BOILER	STEAM CHEST STBD	STEAM CHEST PORT	VACUUM CONDENSER BACK PRESSURE IN AUX. EXH. LINE. (GAGE)	COMPRESSION PRESSURE STARBOARD	COMPRESSION PRESSURE PORT	EXPANSION PRESSURE STARBOARD	EXPANSION PRESSURE PORT	REVOLUTIONS STARBOARD	REVOLUTIONS PORT	TEMP. COLD STOR STARBOARD	TEMP. COLD STOR PORT	CONDENSED EXHAUST STEAM. TIME OF OBSERVATION	INTERVAL	INCHES IN TANK	WEIGHT IN TANK LBS.	TEMPERATURE	WEIGHT PER HR LBS	WEIGHT PER MACHINE	DURATION OF TEST.
TEST NO 1	AM 1 40	230		62	10 6		225		80				26	114000	M S			80			
	1150	240		60	10 6.5		227		80		952		26	121114	3114	10	219	79			
	W 12 00			62			230		80		984		26	124204	3050	20	457	78			
	1210	230		62	10 6		228		80		987		26	11220	3016	30	688	80			
	1220			62			230		80		984		26	14022	2802	40	924	79			
	1230	225		62	7 6		230		80		973		26	20646	2624	FULL	1136	80			
	1240	238		62	8 6		230		80		972		26								
	1250			62			230		80		958		26								
	100	230	running	62	8 6		230	running	80	running	977		26								
	110	235		61	10 6		228		80		965		26								
	120			60			225		80		957		26								
	130	235	Not	62	10 6		227	Not	80		978		26								
	140	230		62	10 6.5		225		80		957		26								
	150			62			225		80		941		26								
	200	230		62	10 7		225		80		938		26								
	210	225		62	10 7		225		80		912		26								
	AVERAGE	232		62	9 62		228		80		962		26						464.4		2 Hr 26 M 46 S.
TEST NO 2		250	245	70	66 10 8	265	225	80	80			245	245	230000				75			
		300	225	70	66 10 6	255	225	80	77	955	961	245	245	300435	1045	10	219	76			
		310	225	70	65 10 6	255	220	77	75	948	955	245	245	314115	1532	20	457	75			
		320	225	68	65 10 6	255	220	77	75	977	969	245	245	326446	1231	30	688	75			
		330	225	68	65 10 6	255	220	77	75	998	983	24	24	339448	1302	40	924	75			
		340	240	70	65 10 65	257	223	77	75	996	978	24	24	351427	1139	FULL	1136	75			
		350	225	70	65 10 6	257	222	77	77	982	987	235	235								
	AVERAGE	230	65	65	10 64	257	222	76	76	982	972	242	242						1110	555	1 Hr 01 M 27 S

TABLE 32

TESTS OF STEERING ENGINES.

NO. OF CYLINDERS = 2.
DIAM. OF CYL'S. = 12".
STROKE = 10"
TYPE, RIGHT & LEFT SCREW.
MAKER { HYDE WINDLASS CO. (CHESTER).
WILLIAMSON BROS. (BIRMINGHAM & SALEM).

OCT. 10-12-1908.

NUMBER OF TEST	VESSEL	STEAM PRESSURE. GAGE.		VACUUM. (INCHES). AUX. CONDENSER.	DURATION OF TEST. H. M. S.	CONDENSED EXHAUST STEAM. (POUNDS)	
		BOILER	AUXILIARY EXHAUST LINE.			TOTAL	PER HOUR.
1	BIRMINGHAM	176	6.	20.6	0-40-10	687	*1026
2	CHESTER	214	6.	10.	1-16-20	689	* 542
3	SALEM	258	6.	10.	1-22-00	672	* 492
4	BIRMINGHAM	168	3.5	20.6	0-45-07	1136	§1511
5	CHESTER	206	6.5	10.	0-52-10	689	§ 793
6	SALEM	258	6.	10.	0-43-53	672	§ 919

* Helm, amidships; steam to engine open but engine not in operation. In these tests the engine, after being thoroughly warmed up, was allowed to remain at rest. The condensed exhaust steam collected, therefore, was that due to leakage alone.

§ The helm in these tests, in order to approximate conditions when under way, was, every two minutes, thrown three degrees to starboard, then the same amount to port, and immediately afterwards brought to mid position.

TABLE T.—*Synopsis of results of auxiliary machinery tests.*

Auxiliary.	No. of table and test.	Duration of test.	Steam-chest pressure gage.	Revolutions or double strokes per minute.	I. H. P. (average).	Condensed exhaust steam. (Pounds.) Per hour.	Per I. H. P. per hour.	Remarks.
		H. m. s.						
Main air pumps	2-1	1 30 21	50.0	26.90	4.0	753.1	188.2	1 cylinder only in use.
	2-2	1 31 22	44.0	21.10	3.5	744.8	212.8	Do.
	2-3	2 09 40	42.0	14.30	2.8	524.7	187.4	Do.
	2-4	0 40 46	48.0	26.90	7.3	1,663.1	227.8	Both cylinders in use.
	3-1	0 35 39	105.0	28.90	21.0	3,824.1	182.1	Vacuum main condenser, 23.1.
	3-2	0 28 52	82.0	21.90	16.9	2,361.0	139.7	Vacuum main condenser, 21.0.
	3-3	0 42 49	73.0	15.90	14.1	1,593.2	113.0	Vacuum main condenser, 18.1.
	3-4	1 15 43	58.0	9.60	7.8	900.9	115.5	Vacuum main condenser, 14.0.
Turbine wet vacuum pumps	4-1	2 13 40	199.0	1,314.00	.....	508.1	.....	1 nozzle open to steam turbine.
	4-2	1 35 21	150.0	1,292.00	.....	712.3	.....	2 nozzles open to steam turbine.
Dry vacuum pumps	5-1	0 49 30	85.0	60.10	20.7	1,373.3	66.3	
	5-2	0 42 05	71.0	80.10	26.1	1,615.4	61.89	
	5-3	0 59 48	70.0	45.20	14.2	1,136.8	80.0	
Main circulating pumps	6-1	0 41 25	104.4	239.00	48.9	1,637.0	33.5	Main injection and discharge valves wide open during all tests.
	6-2	1 18 33	59.3	177.00	21.5	866.2	40.3	
	6-3	2 23 15	32.2	110.00	6.0	416.7	69.4	
	6-4	2 36 00	15.7	58.00	1.8	219.3	121.8	
	7-1	1 52 14	46.0	87.50	3.6	302.6	84.1	
	7-2	1 24 33	82.0	155.10	15.9	795.5	50.0	
	7-3	0 25 20	173.0	275.80	81.0	2,643.0	32.6	
	7-4	0 16 37	193.0	406.20	161.6	3,990.0	24.7	
	8-1	0 39 12	128.0	233.20	142.9	3,482.5	24.35	
	8-2	0 35 28	100.0	186.90	87.0	1,922.0	22.0	
	8-3	1 00 20	73.0	145.00	37.0	1,135.0	30.6	
	8-4	1 57 55	35.0	76.80	9.3	411.1	44.2	

TABLE T.—*Synopsis of results of auxiliary machinery tests*—Continued.

Auxiliary	No. of table and test.	Duration of test.	Steam-chest pressure gage.	Revolutions or double strokes per minute.	I. H. P. (average).	Condensed exhaust steam. (Pounds.) Per hour.	Per I. H. P. per hour.	Remarks.
		H. m. s.						
Main feed pumps	9-1	0 14 32	198.0	50.00	66.4	4,661.0	70.2	Discharge pressure gage, 200 pounds.
	9-2	0 21 10	183.0	30.00	46.4	3,214.5	69.3	Discharge pressure gage, 212 pounds.
	10-1	0 37 47	165.0	12.80	19.8	1,775.3	89.7	Discharge pressure gage, 300 pounds.
	10-2	0 22 57	203.0	24.80	41.7	2,917.0	70.0	Do.
	10-3	0 40 00	211.0	34.30	52.8	4,185.6	79.4	Do.
	10-4	0 24 30	240.0	45.30	72.1	5,446.7	74.55	Do.
	10-5	0 17 12	223.0	34.30	52.4	3,934.9	75.1	Do.
	11-1	0 28 37	153.0	13.00	18.1	2,381.7	131.6	Discharge pressure gage, 268 pounds.
	11-2	0 21 02	187.0	24.50	41.7	3,229.4	77.4	Discharge pressure gage, 265 pounds.
	11-3	0 28 30	254.0	33.00	59.6	4,766.4	80.0	Discharge pressure gage, 286 pounds.
Engine-room fire and bilge pumps	12-1	1 08 20	75.0	55.00	9.5	999.2	105.2	Discharge pressure gage, 30 pounds.
	12-2	0 35 55	99.0	53.20	11.1	1,147.7	103.4	Discharge pressure gage, 45 pounds.
	12-3	1 10 56	46.0	54.00	5.6	581.1	103.8	Discharge pressure gage, 15 pounds.
	13-1	1 41 10	49.0	41.90	2.6	398.5	153.3	Discharge pressure gage, 10 pounds.
	13-2	1 49 50	56.0	39.70	3.8	612.4	161.2	Discharge pressure gage, 20 pounds.
	13-3	1 34 05	87.0	40.30	7.0	714.9	102.1	Discharge pressure gage, 40 pounds
	14-1	0 39 12	120.0	54.90	13.3	1,740.2	130.8	Do.
	14-2	1 24 32	85.0	56.10	8.3	807.1	97.2	Discharge pressure gage, 20 pounds
	14-3	1 05 12	56.0	40.30	4.1	420.6	102.5	Do.
	14-4	0 45 36	79.0	55.90	6.6	755.2	114.4	Discharge pressure gage, 15 pounds
Engine-room auxiliary air and circulating pump.	15-1	1 35 11	67.0	60.00	.....	504.9	.....	Power not indicated.
	15-1	1 39 00	.....	44.80	1.6	266.1	166.3	
	16-1	1 22 35	100.0	59.90	4.7	500.5	106.5	

Fireroom auxiliary feed pumps	17-1	1 45 44	156.0	13.30	4.8	643.5	134.1	Discharge pressure gage, 311 pounds.
	17-1	0 35 10	258.0	35.00	20.3	1,907.5	94.0	Discharge pressure gage, 300 pounds.
	17-2	0 47 40	225.0	25.00	14.3	1,409.8	98.6	Do.
	17-3	0 15 15	190.0	15.00	7.8	893.8	114.6	Do.
	18-1	0 52 39	187.0	29.70	14.7	1,295.7	88.14	Do.
Fire and bilge on ash ejector	19-1	0 32 26	180.0	36.00	25.6	2,101.6	82.1	Discharge pressure gage, 280 pounds.
	19-1	0 15 58	263.0	50.00	40.2	4,193.6	104.3	Discharge pressure gage, 265 pounds.
	19-2	0 24 25	202.0	36.60	25.0	2,742.3	109.7	Do.
	19-3	0 40 55	166.0	22.20	13.9	1,639.4	117.9	Do.
	20-1	0 25 30	176.0	36.30	31.4	2,674.0	85.1	Discharge pressure gage, 262 pounds.
Forced-draft blowers	21-1	1 18 30	125.0	380.00	.....	1,738.7	.....	No air pressure in fireroom.
	21-2	0 43 30	110.0	360.00	.....	1,569.7	.....	Do.
	21-3	1 06 15	95.0	310.00	.....	1,030.6	.....	Do.
	21-4	0 45 00	75.0	260.00	.....	758.7	.....	Do.
	21-5	0 36 50	124.0	379.00	.....	1,853.8	.....	Do.
	22-1	0 12 06	149.5	486.00	.....	5,509.0	.....	4 inches of air pressure.
	22-2	0 17 34	106.0	399.50	32.2	3,811.6	59.2	2½ inches of air pressure.
	22-3	0 53 31	54.0	223.50	6.7	1,255.7	93.7	1 inch of air pressure.
	22-4	0 53 44	41.5	165.00	3.2	753.7	117.8	½ inch of air pressure.
	22-5	0 13 43	125.0	427.00	35.9	4,872.9	67.9	3 inches of air pressure.
	22-6	0 28 00	76.5	305.00	16.6	2,391.5	72.0	1½ inches of air pressure.
	23-1	0 26 10	106.0	393.00	21.7	2,004.8	60.0	4 inches of air pressure.
	23-2	0 41 46	70.0	313.00	11.7	1,631.9	69.7	2½ inches of air pressure.
	23-3	1 05 03	42.0	190.00	3.3	636.4	96.4	1 inch of air pressure.
	23-4	1 08 03	52.1	206.00	4.7	1,001.6	106.4	Do.
	23-5	0 32 09	87.0	327.00	15.9	2,120.2	66.7	2½ inches of air pressure.
	23-6	0 19 15	143.0	424.00	34.4	3,534.5	51.4	3.9 inches of air pressure.
	23-7	0 56 28	65.0	223.00	6.7	1,208.2	90.2	1 inch of air pressure.
	23-8	0 21 15	130.0	382.00	27.6	3,204.6	58.1	2½ inches of air pressure.

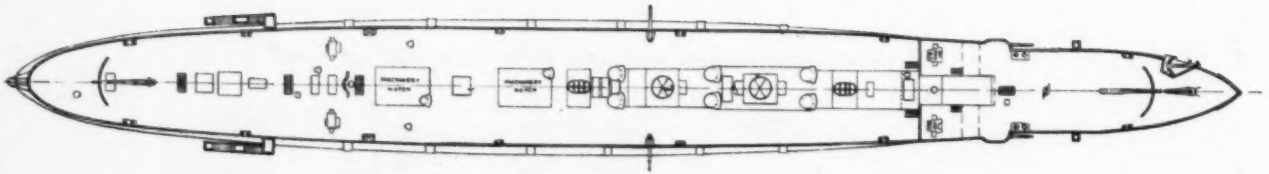
TABLE T.—*Synopsis of results of auxiliary machinery tests*—Continued.

Auxiliary.	No. of table and test.	Duration of test.	Steam-chest pressure gage.	Revolutions or double strokes per minute.	I. H. P. (average).	Condensed exhaust steam. (Pounds.)		Remarks.
						Per hour.	Per I. H. P. per hour.	
		H. m. s.						
Distiller circulating pumps (used for flushing).	24-1	2 55 00	47.0	30.70	3.5	779.0	222.57	Discharge pressure gage, 25 pounds.
	25-1	1 01 47	73.0	53.30	9.5	1,088.6	114.6	Discharge pressure gage, 30 pounds.
	25-2	1 05 38	63.0	35.30	5.9	827.3	140.2	Do.
	25-3	1 11 30	45.0	20.40	3.1	368.4	118.8	Results of test unreliable.
	25-4	1 10 07	88.0	33.00	7.6	959.3	126.2	Discharge pressure gage, 50 pounds.
	25-5	1 39 52	45.0	34.50	3.8	673.5	177.2	Discharge pressure gage, 20 pounds.
	24-1	1 51 50	62.0	54.60	6.1	610.0	100.0	Discharge pressure gage, 12 pounds.
Evaporators and distilling plant	26-1	4 00 00	.....	.....	.....	[a]3,190.8	[b]2,608.3	Two evaporators in use.
	26-1	4 00 00	.....	.....	.....	[a]4,140.0	[b]2,980.3	Do.
	26-1	4 00 00	.....	.....	.....	[a]4,999.2	[b]3,687.8	Do.
Dynamos	27-1	1 08 45	108.0	400.00	20.4	989.6	48.5	
	27-2	1 02 40	107.0	400.00	29.9	1,085.7	36.3	
	27-3	0 52 04	108.0	400.00	39.2	1,306.7	33.3	
	27-4	0 43 40	108.0	400.00	46.6	1,558.1	33.4	I. H. P. figured from similar tests on the Chester.
	27-5	0 34 00	109.0	395.00	55.7	1,994.1	35.8	
	27-6	0 42 41	108.0	400.00	20.8	1,594.0	38.3	
	27-7	1 23 03	108.0	400.00	20.6	819.3	39.8	
	28-1	1 46 50	110.0	400.00	20.4	629.0	30.8	
	28-2	1 26 00	110.0	400.00	29.9	781.4	26.1	
	28-3	1 02 08	111.0	400.00	39.2	1,081.5	27.6	Do.
	28-4	0 51 18	112.0	400.00	46.6	1,310.0	28.1	
	28-5	0 46 28	112.0	400.00	55.7	1,446.2	26.0	
	29-1	0 53 05	100.0	382.00	48.5	1,285.5	26.5	

	29-2	0 44 43	100.0	381.00	53.8	1,525.6	28.4	
	29-3	0 56 52	100.0	386.00	40.5	1,207.7	29.8	
	29-4	2 20 30	96.0	388.00	20.4	485.6	23.8	Steam consumption unreliable
	29-5	1 50 35	100.0	386.00	30.1	616.9	20.5	Do.
	29-6	1 17 55	101.0	387.00	.....	875.5	29.1	
	29-7	1 43 27	103.0	393.00	.....	650.4	32.3	
	30-1	0 50 55	73.5	118.00	.....	670.5	.....	Power not indicated.
Ice machines	30-1	1 09 08	49.0	95.85	.....	486.1	.....	Do.
	31-1	2 26 46	62.0	96.20	.....	464.4	.....	Do.
	31-2	1 01 27	67.0	97.70	.....	555.0	.....	Do.
	32-1	0 40 10	.....	.....	.....	1,026.0	.....	Engine at rest.
	32-4	0 45 07	.....	.....	.....	1 511.0	.....	
Steering engines	32-3	1 22 00	.....	.....	.....	492.0	.....	Do.
	32-6	0 43 53	.....	.....	.....	919.0	.....	
	32-2	1 16 20	.....	.....	.....	542.0	.....	Do.
	32-5	0 52 10	.....	.....	.....	793.0	.....	

[a] Total steam used per hour by entire plant.

[b] Pounds of water distilled per hour.



Length between perpendiculars	420 Ft 0"
Length on L.W.L.	420 Ft 0"
Length over all	423 Ft 2"
Breadth, moulded	46 Ft 8"
Breadth, extreme	47 Ft 0⅜"
Ratio of Length to Beam	8.97
Normal Displacement (16 Ft 9¼ Draft)	3780 Tons.

U.S.S. BIRMINGHAM & SALEM

PROFILE INBOARD

SCALE ⅛" = 1 Ft.

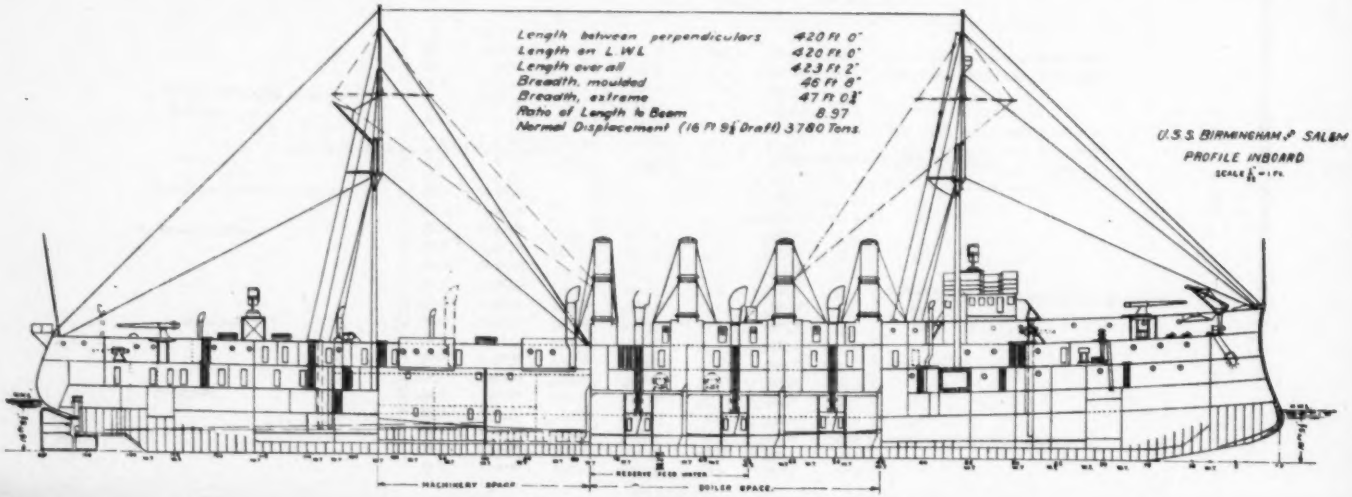
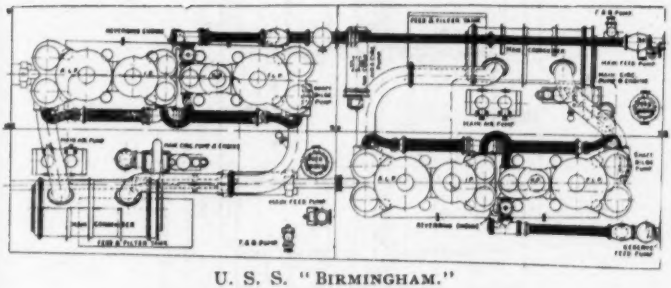
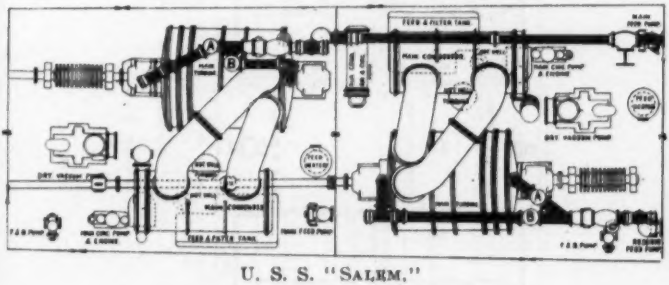


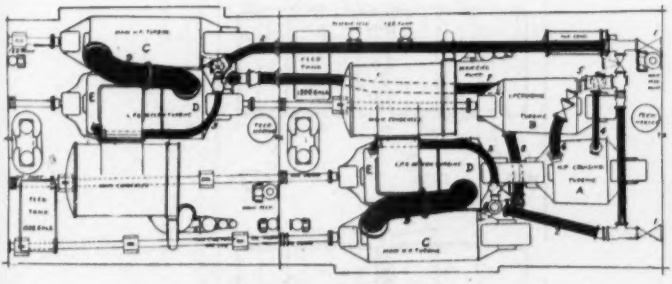
Plate I.



U. S. S. "BIRMINGHAM."



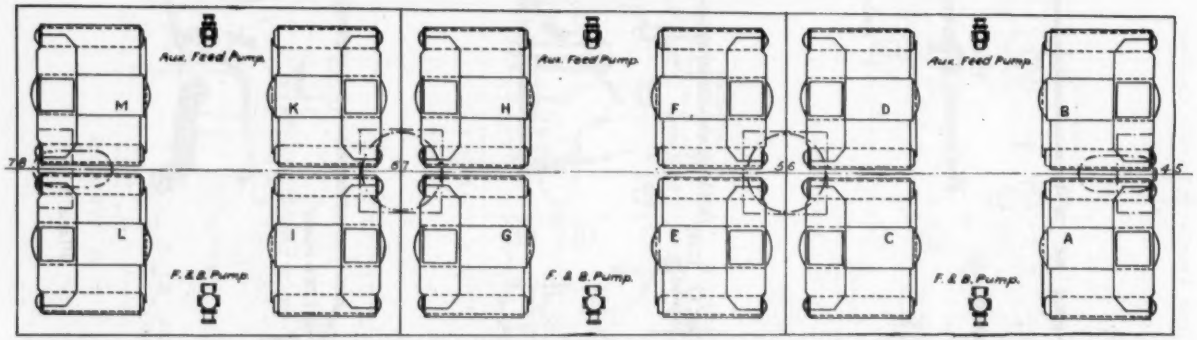
U. S. S. "SALEM."



U. S. S. "CHESTER."

Plate 2.

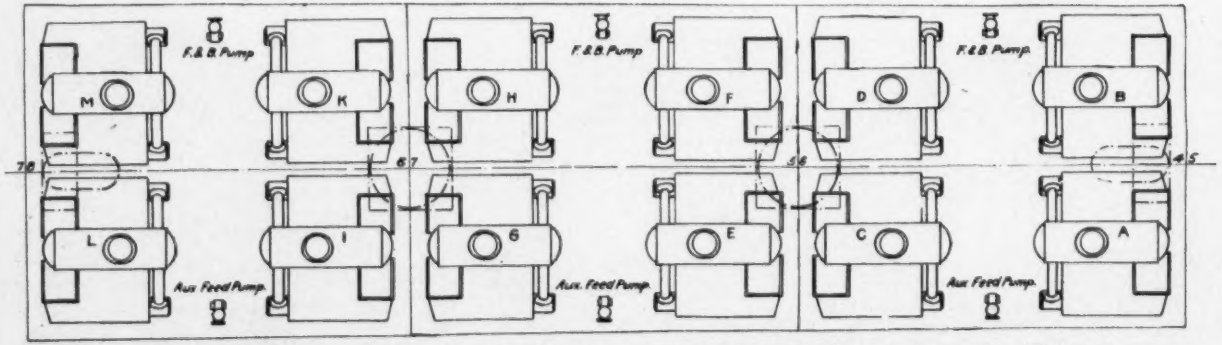
Plate I.



Twelve Fore River water-tube Express boilers. Grate surface, one boiler, 58 sq. ft. Total grate surface, 696 sq. ft. Heating surface, one boiler, 3,166 sq. ft. Total heating surface, 37,992 sq. ft. Ratio total G. S. to total H. S., 1: 54.6. Working pressure, 250 pounds.

U. S. S. "SALEM" AND U. S. S. "BIRMINGHAM."—PLAN OF FIREROOMS.

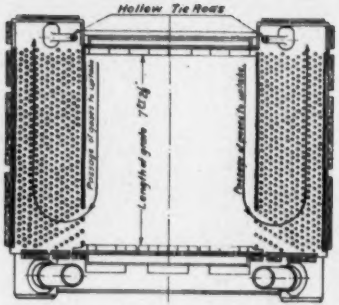
Plate 3.



Twelve Normand water-tube boilers. Grate surface, one boiler, 58 sq. ft. Total grate surface, 696 sq. ft. Heating surface, one boiler, 2,670 sq. ft. Total heating surface, 32,040 sq. ft. Ratio total G. S. to total H. S., 1 : 46.03. Working pressure, 250 pounds.

U. S. S. "CHESTER."—PLAN OF FIREROOMS.

Plate 3.

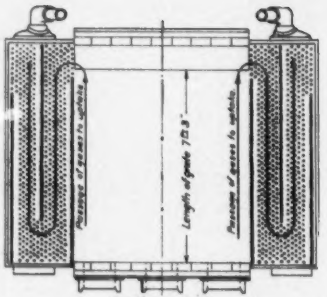


Section on A-B

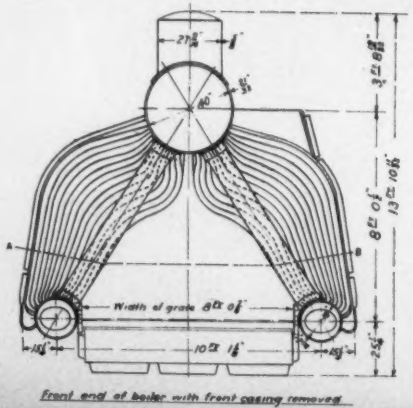
U.S.S CHESTER

LAY-OUT OF BOILER TUBES.

Number of tubes	986
Outside dia of tubes	1¾"
Thickness of tubes	#10 B.W.G.
Heating surface	2670 sq.ft
Grate surface	58 sq.ft.
Ratio G.S. to H.S.	1 : 46.03
Working pressure	250 Pounds.



Section on C-D



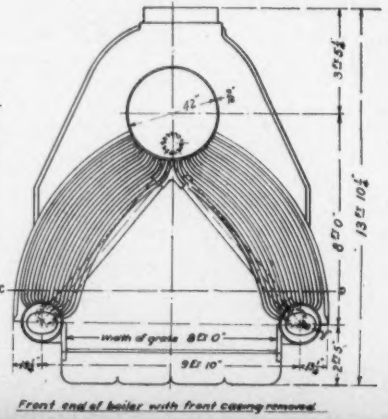
Front end of boiler with front casing removed

U.S.S. BIRMINGHAM.

U.S.S SALEM

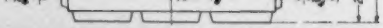
LAY-OUT OF BOILER TUBES.

Number of tubes	1462
Outside dia of tubes	1¾"
Thickness of tubes	#11 B.W.G.
Heating surface	3166 sq.ft.
Grate surface	58 sq.ft.
Ratio G.S. to H.S.	1 : 54.6
Working pressure	250 Pounds

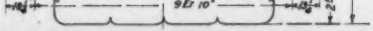


Front end of boiler with front casing removed

Plate 4.



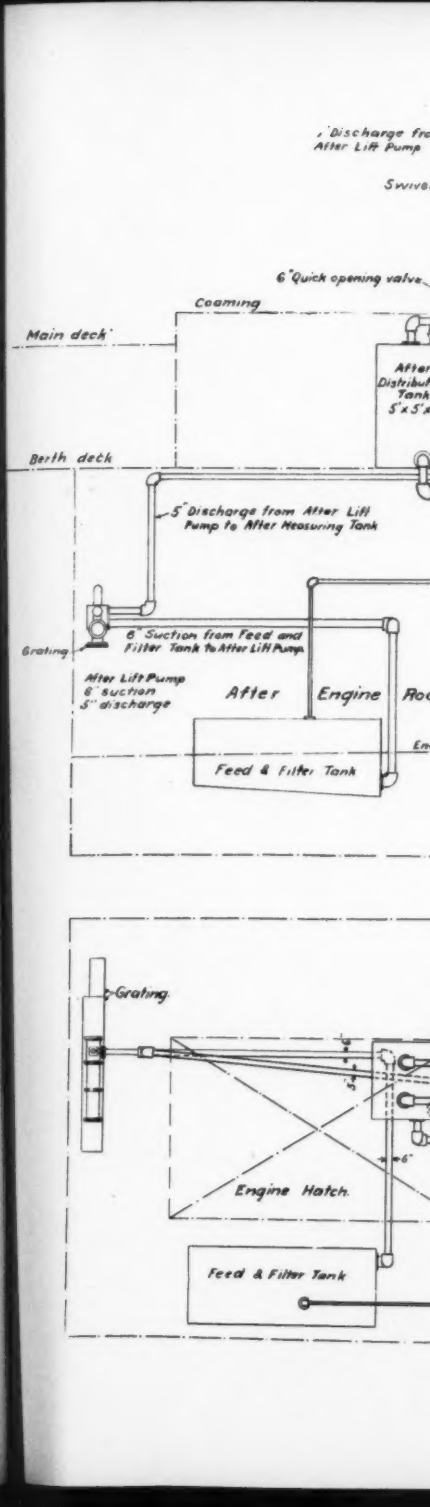
Front end of boiler with front casing removed



Front end of boiler with front casing removed

Plate 4.

Plate 4.


Discharge fro
After Lift Pump
Swivel
6" Quick opening valve
Coaming
Main deck
After
Distribut
Tank
5'x5'x
Berth deck
5" Discharge from After Lift
Pump to After Measuring Tank
6" Suction from Feed and
Filter Tank to After Lift Pump
Grating
After Lift Pump
6" suction
5" discharge
After
Engine
Ro
Em
Feed & Filter Tank
Grating
Engine Hatch.
Feed & Filter Tank

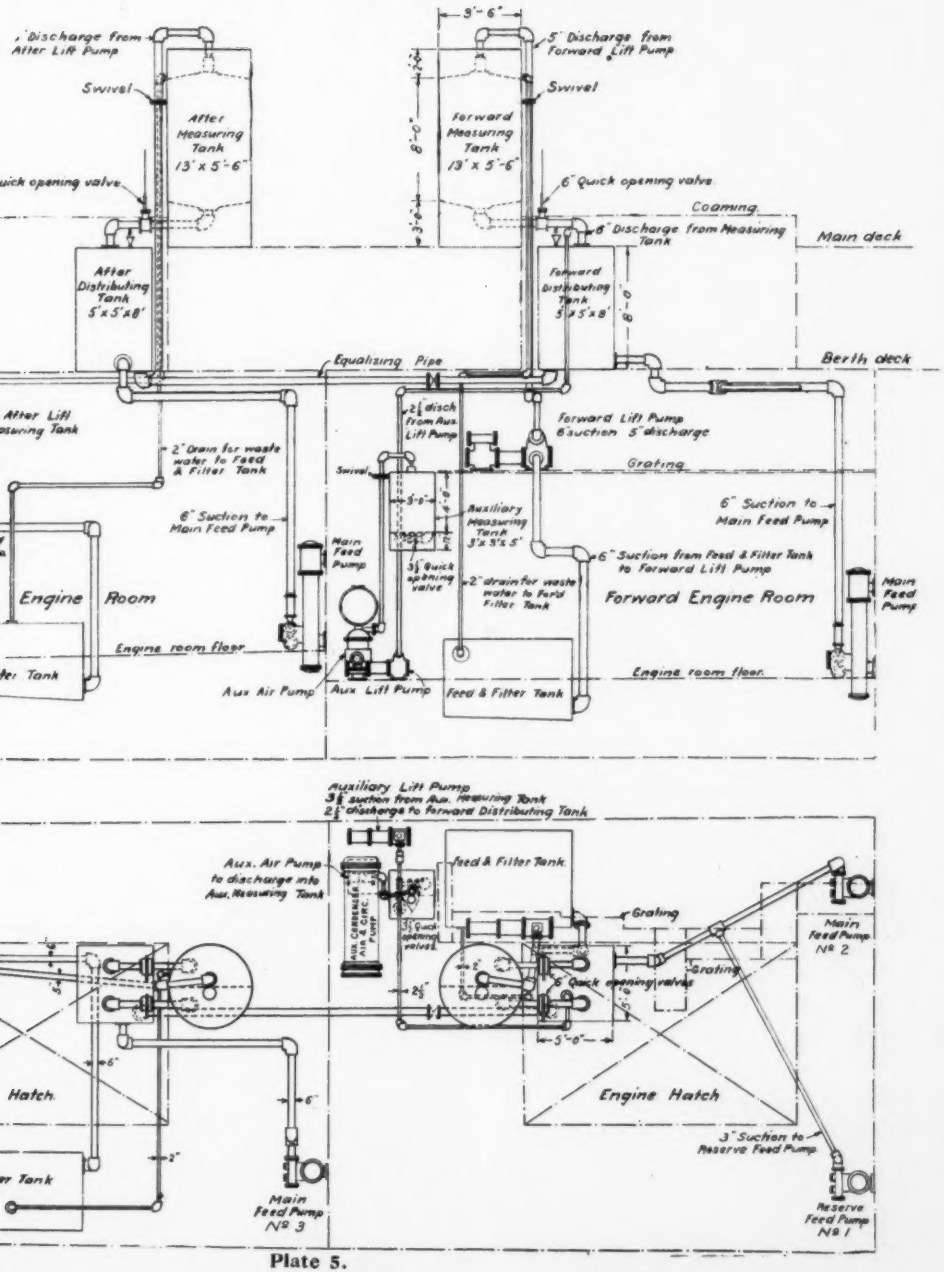

Discharge from After Lift Pump
Swivel
After Measuring Tank 13' x 5'-6"
Quick opening valve
After Distributing Tank 5'x5'x8'
3'-6"
5" Discharge from Forward Lift Pump
Swivel
Forward Measuring Tank 13' x 5'-6"
6" Quick opening valve
Coaming
8" Discharge from Measuring Tank
Main deck
Forward Distributing Tank 5'x5'x8'
8'-0"
Equalizing Pipe
Berth deck
2" disch from Aux Lift Pump
Forward Lift Pump 6" suction 5" discharge
Grating
2" Drain for waste water to Feed & Filter Tank
6" Suction to Main Feed Pump
Swivel
3'-6"
Auxiliary Measuring Tank 3'x3'x5'
3½ Quick opening valve
Main Feed Pump
6" Suction to Main Feed Pump
6" Suction from Feed & Filter Tank to Forward Lift Pump
2" drain for waste water to Feed Filter Tank
Forward Engine Room
Main Feed Pump
Engine Room
Engine room floor
Engine room floor
Aux Air Pump
Aux Lift Pump
Feed & Filter Tank
Auxiliary Lift Pump
3½ Suction from Aux. Measuring Tank
2½ discharge to Forward Distributing Tank
Aux. Air Pump to discharge into Aux. Measuring Tank
Feed & Filter Tank
Grating
Main Feed Pump No 2
3½ Quick opening valves
Grating
2½
5 Quick opening valves
5'-0"
Engine Hatch
Hatch
6"
3" Suction to Reserve Feed Pump
Main Feed Pump No 3
Reserve Feed Pump No 1

Plate 5.

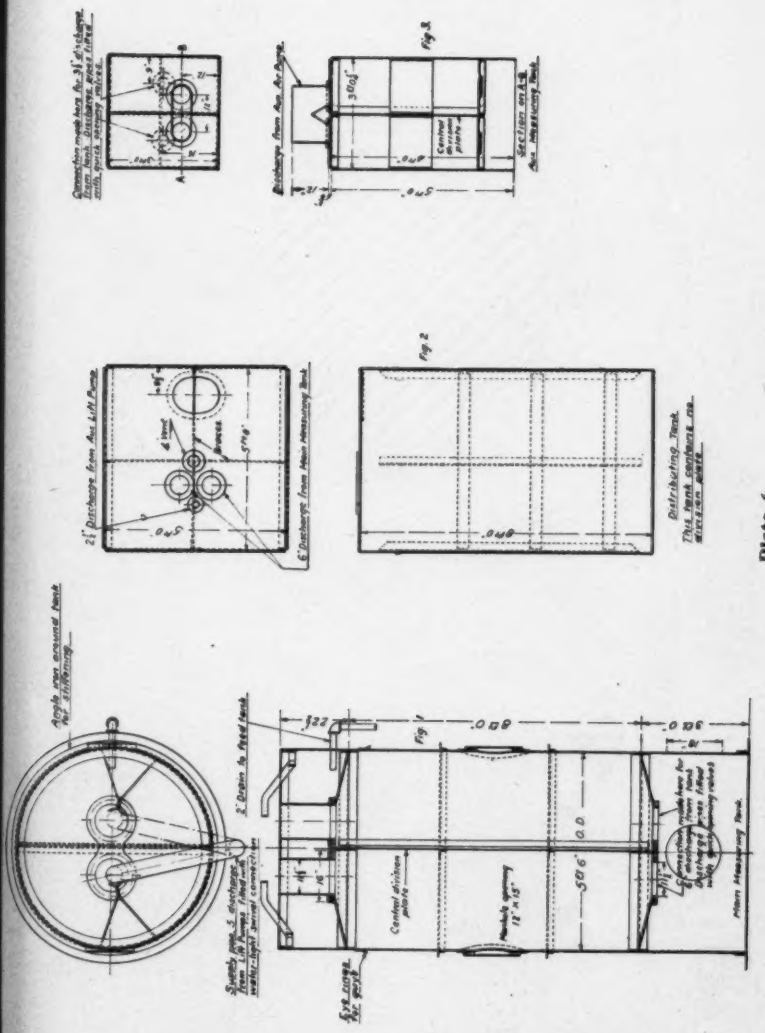


Plate 6.

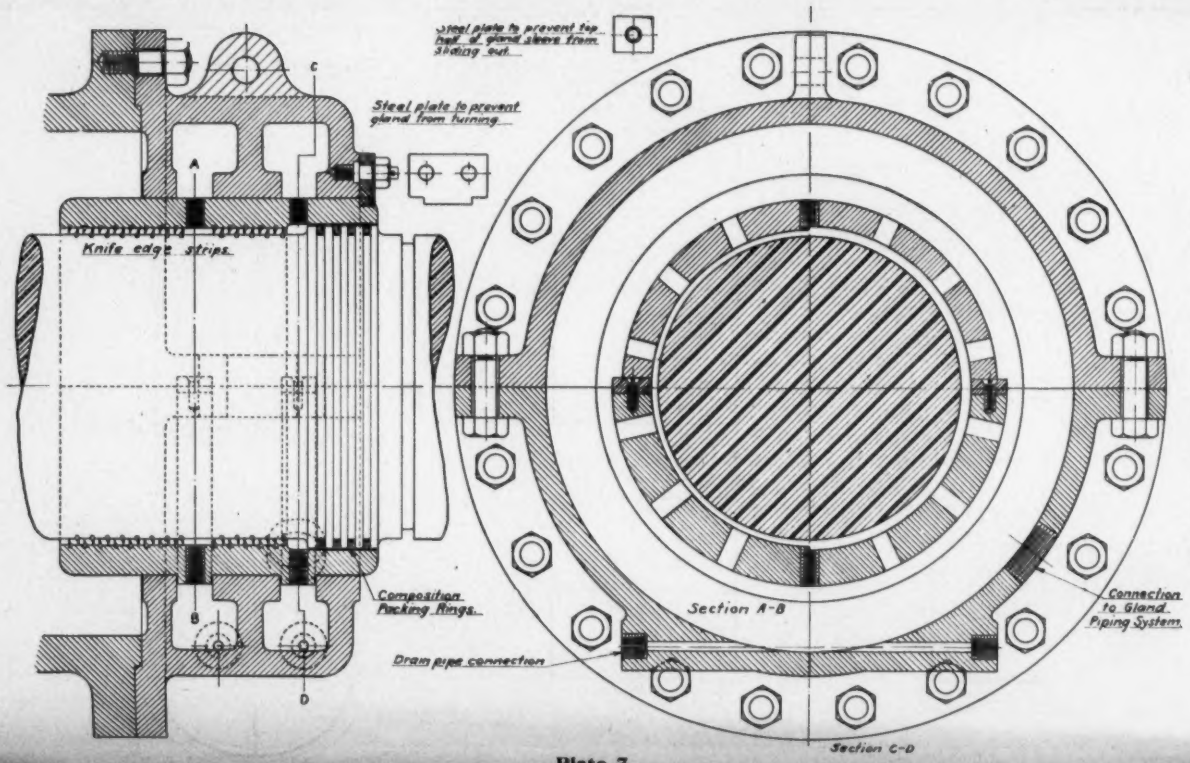

Steel plate to prevent top half of gland sleeve from sliding out.
Steel plate to prevent gland from turning.
Knife edge strips.
Composition Packing Rings.
Drain pipe connection
Section A-B
Connection to Gland Piping System.
Section C-D
A
B
C
D

Plate 7.

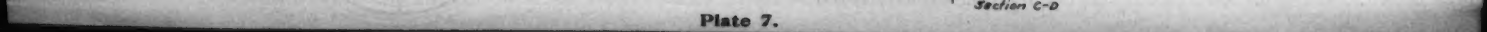


Plate 7.

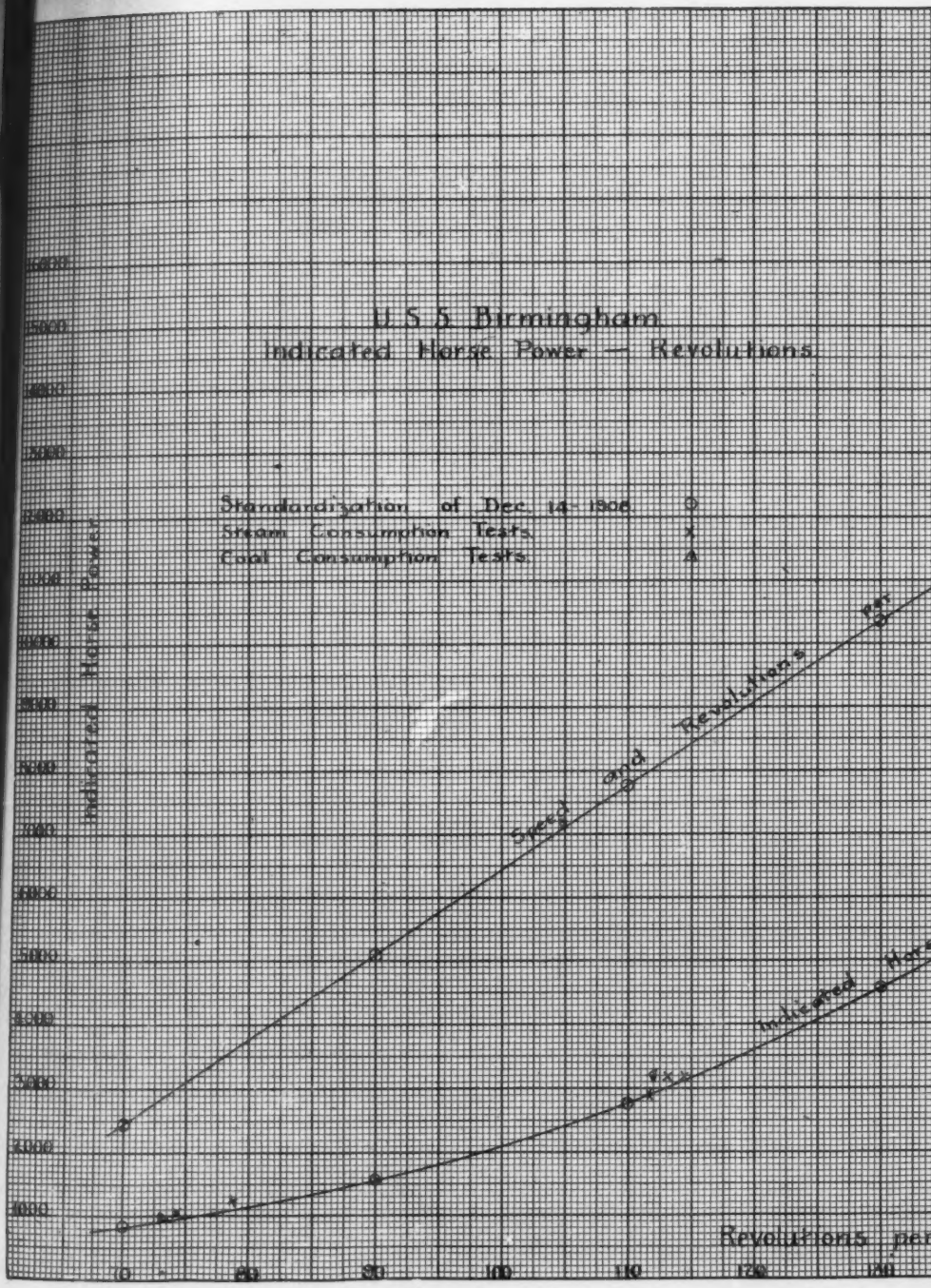


Plate 7.

Section C-D

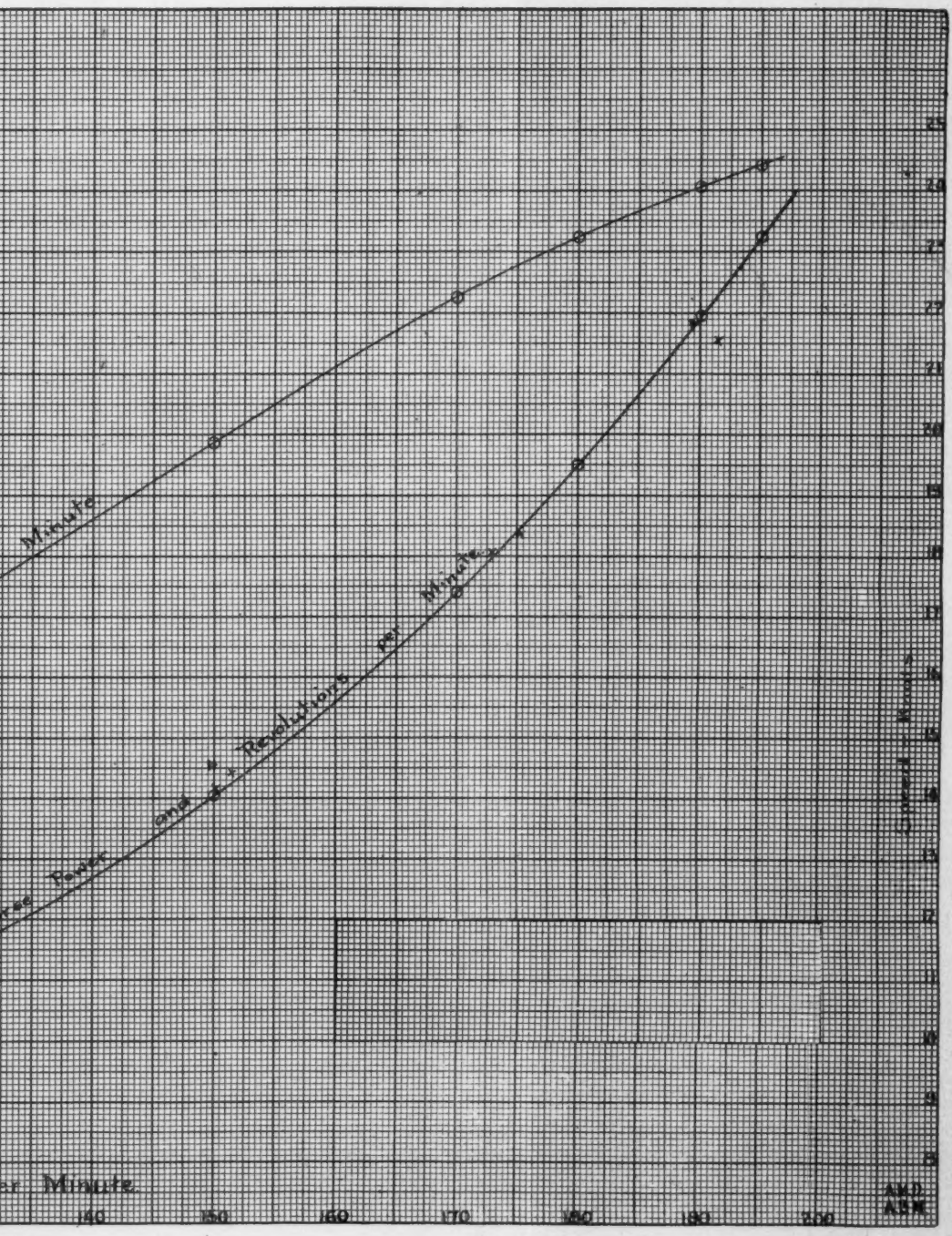
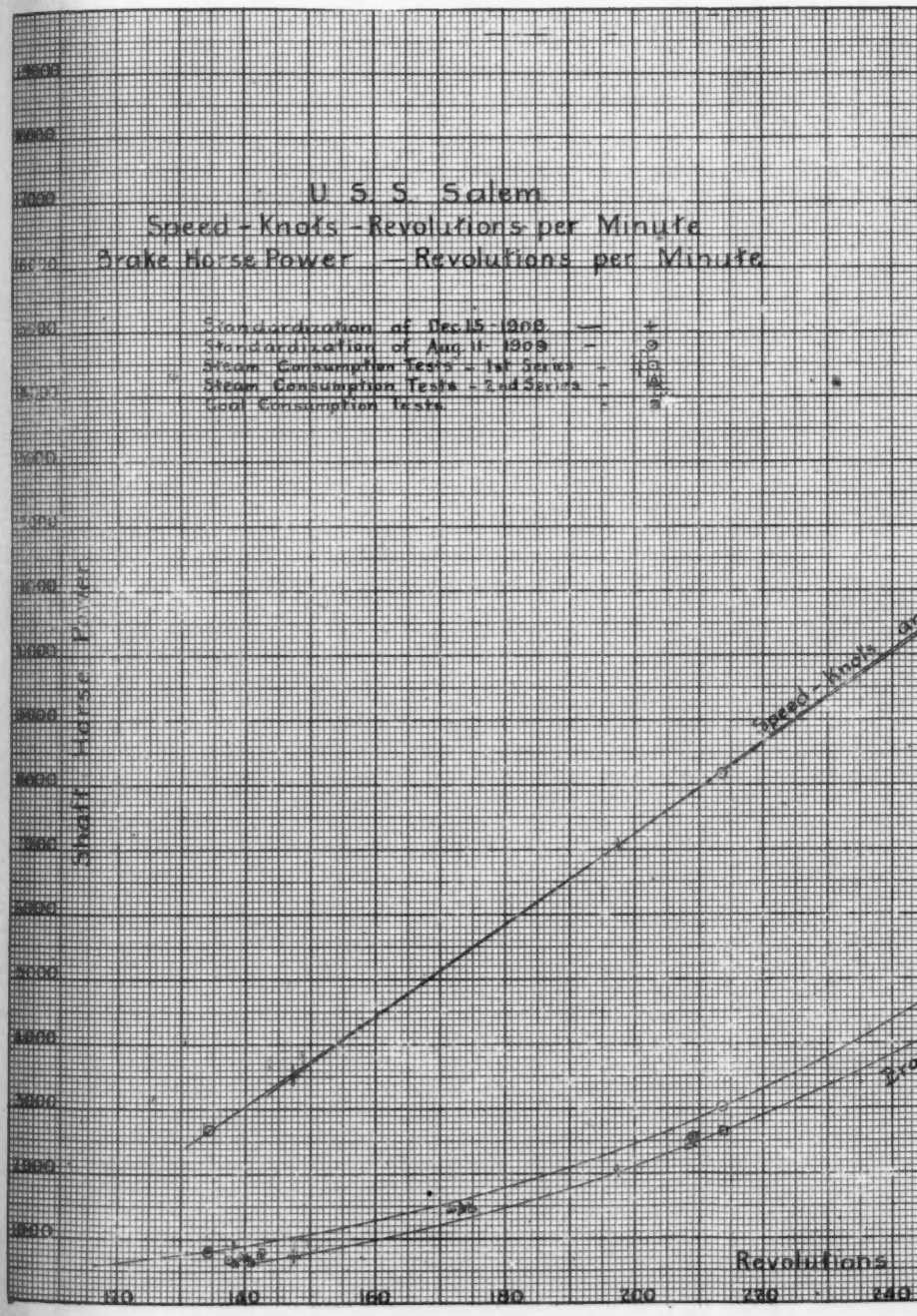

Minute
Power and Revolutions per Minute
Indicated Horse
Minute.
140
150
160
170
180
200
A.W.D.
A.S.M.

Plate 8.


U. S. S. Salem
Speed - Knots - Revolutions per Minute
Brake Horse Power — Revolutions per Minute
Standardization of Dec 15 - 1908
Standardization of Aug 11 1909
Steam Consumption Tests - 1st Series
Steam Consumption Tests - 2nd Series
Coal Consumption Tests
Shaft Horse Power
Speed - Knots
Revolutions
120
140
160
180
200
220
240

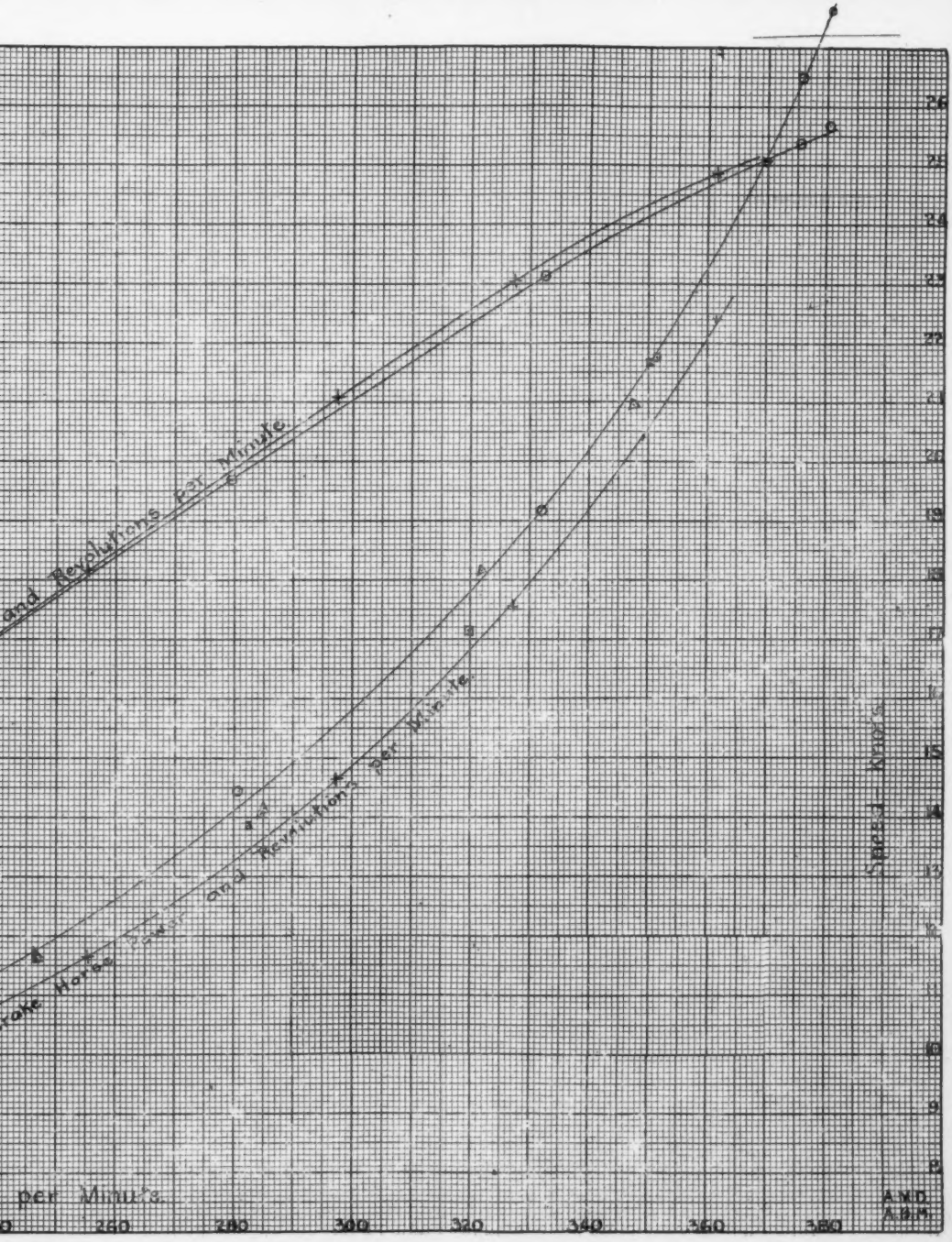
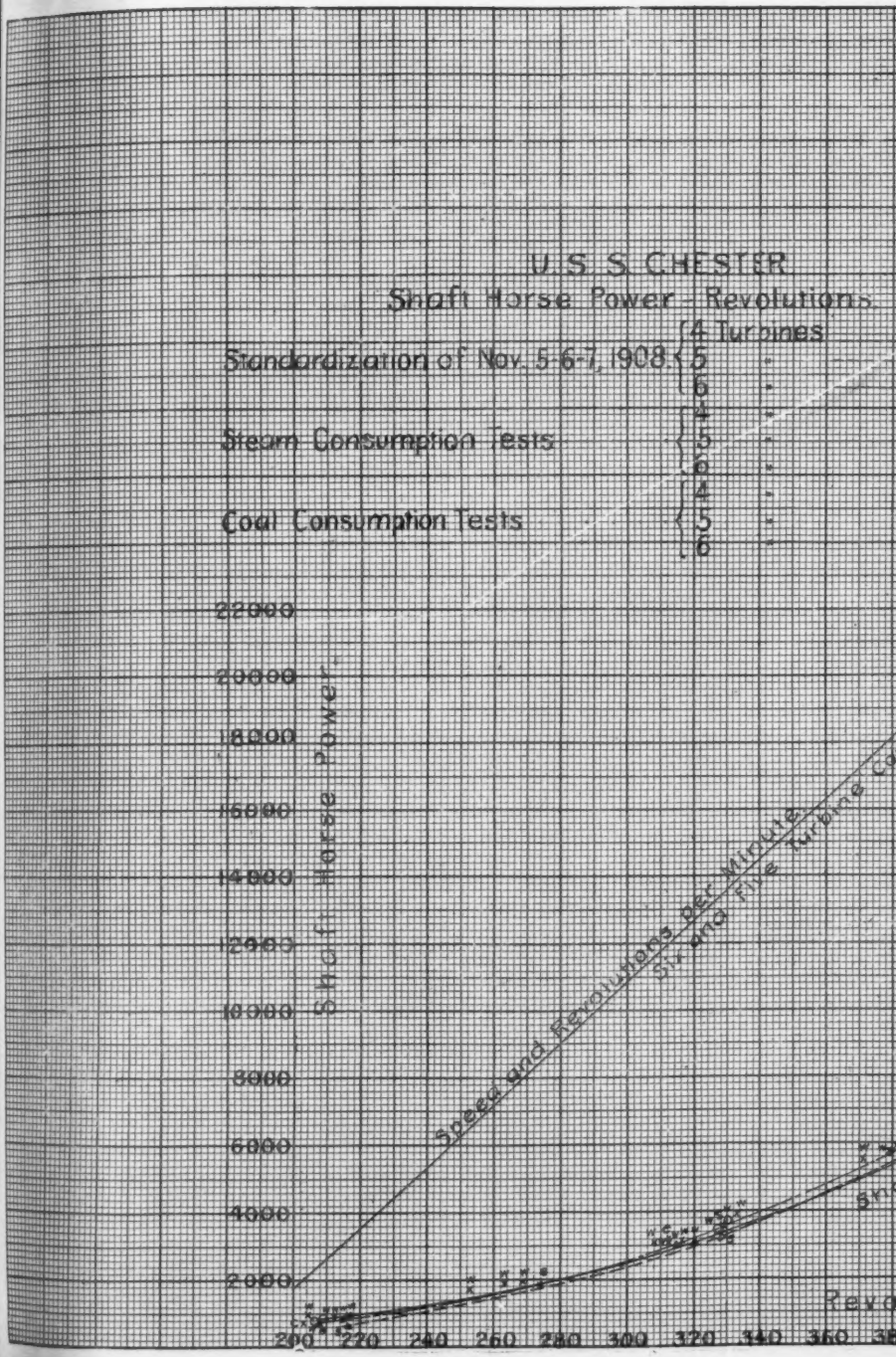

and Revolutions per Minute
rake Horse
Power and Revolutions per Minute
Speed Knots
per Minute
260
280
300
320
340
360
380

Plate 9.


U.S.S. CHESTER
Shaft Horse Power - Revolutions
Standardization of Nov. 5-6-7, 1908.
Steam Consumption Tests
Coal Consumption Tests
4 Turbines
5
6
4
5
6
4
5
6
22000
20000
18000
16000
14000
12000
10000
8000
6000
4000
2000
Shaft Horse Power
Speed and Revolutions per Minute
Six and Five Turbine
260
270
280
290
300
310
320
330
340
350
Revo

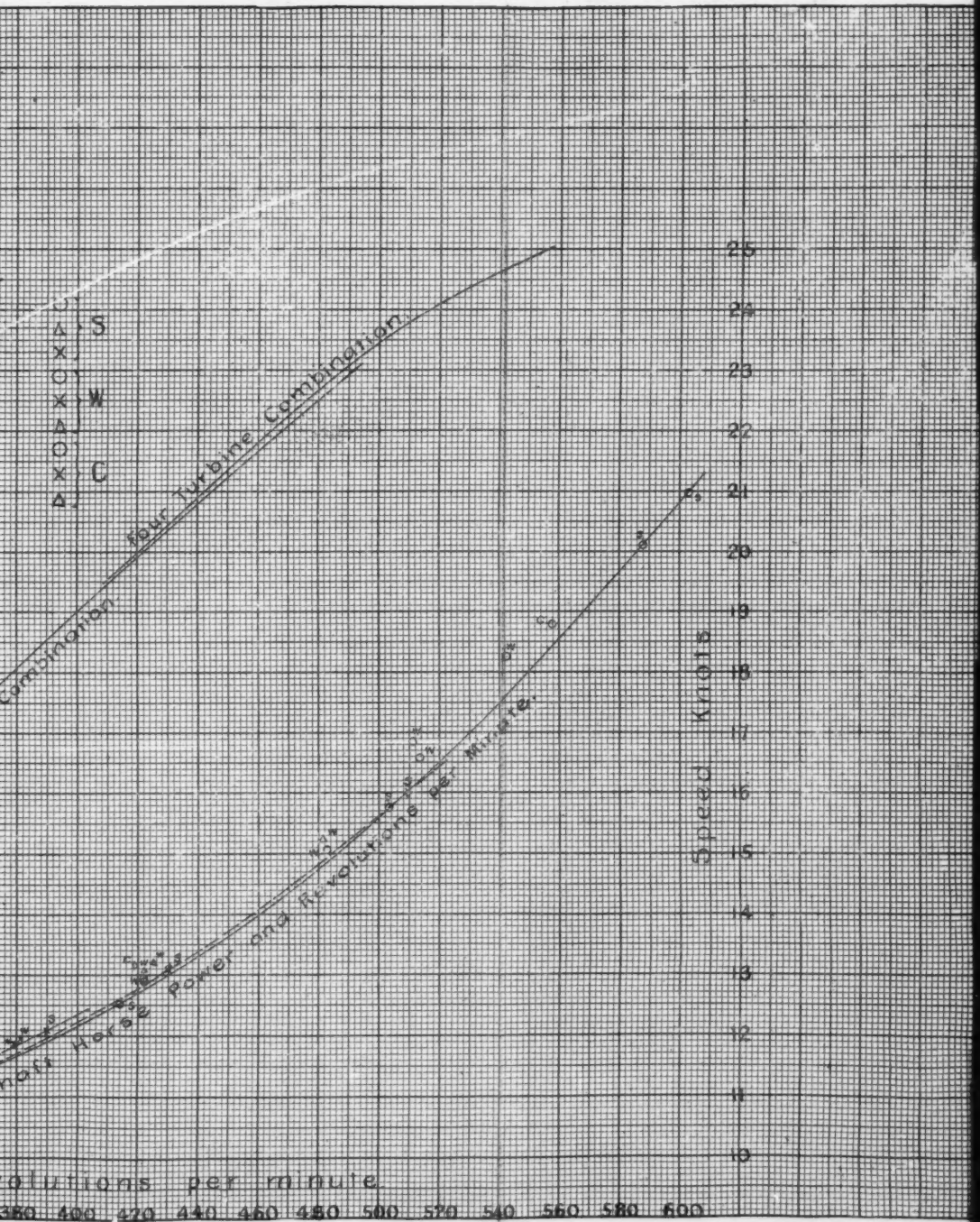
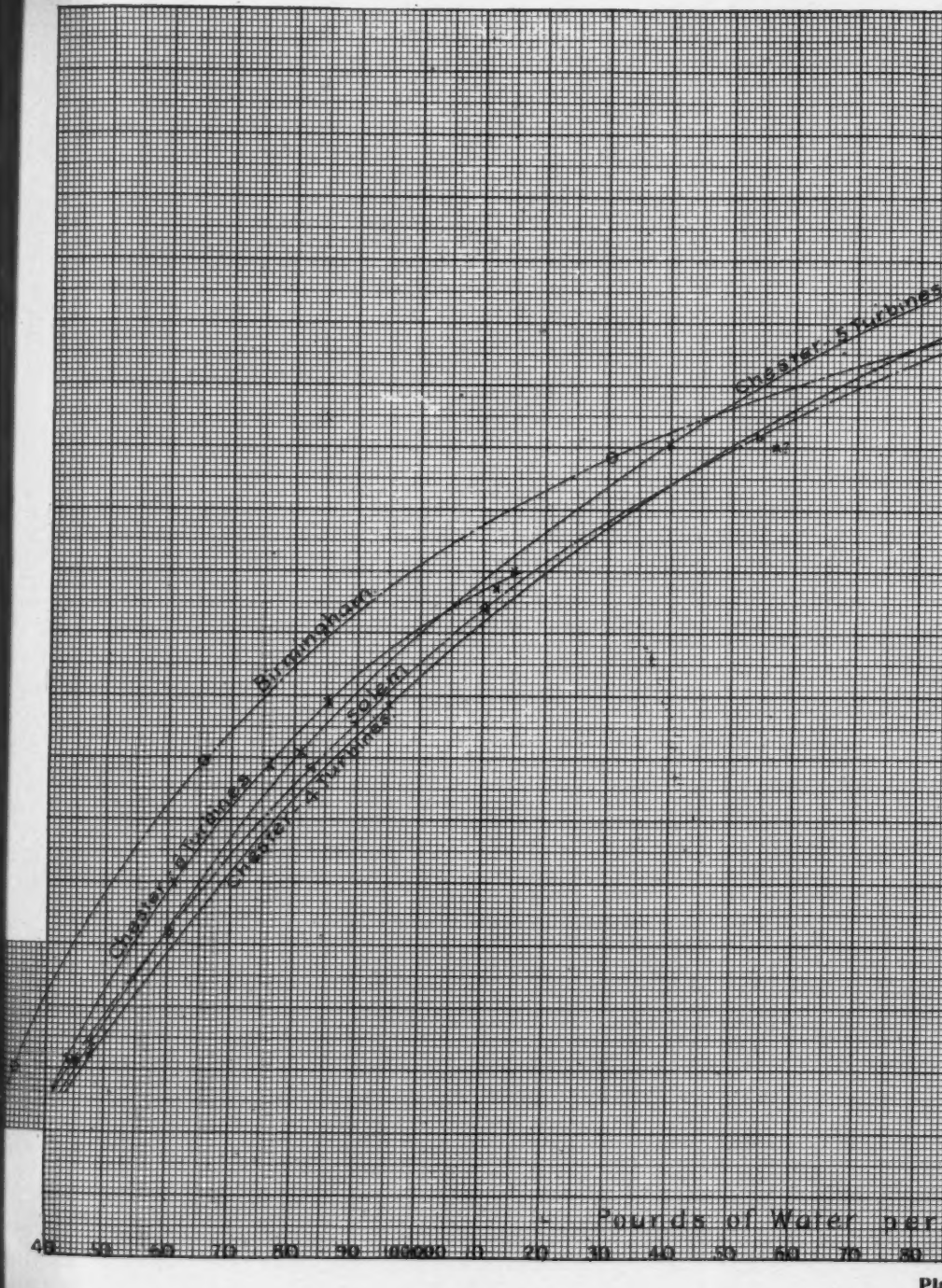

Four Turbine Combination.
Three Turbine Combination.
Power and Revolutions per Minute.
Speed Knots
revolutions per minute.
350 400 420 440 460 480 500 520 540 560 580 600
10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25

Plate 10.


Chester 5 Turbines
Birmingham
Chester 4 Turbines
Soleil
Chester 3 Turbines
Pounds of Water per
40
50
60
70
80
90
100000
0
20
30
40
50
60
70
80

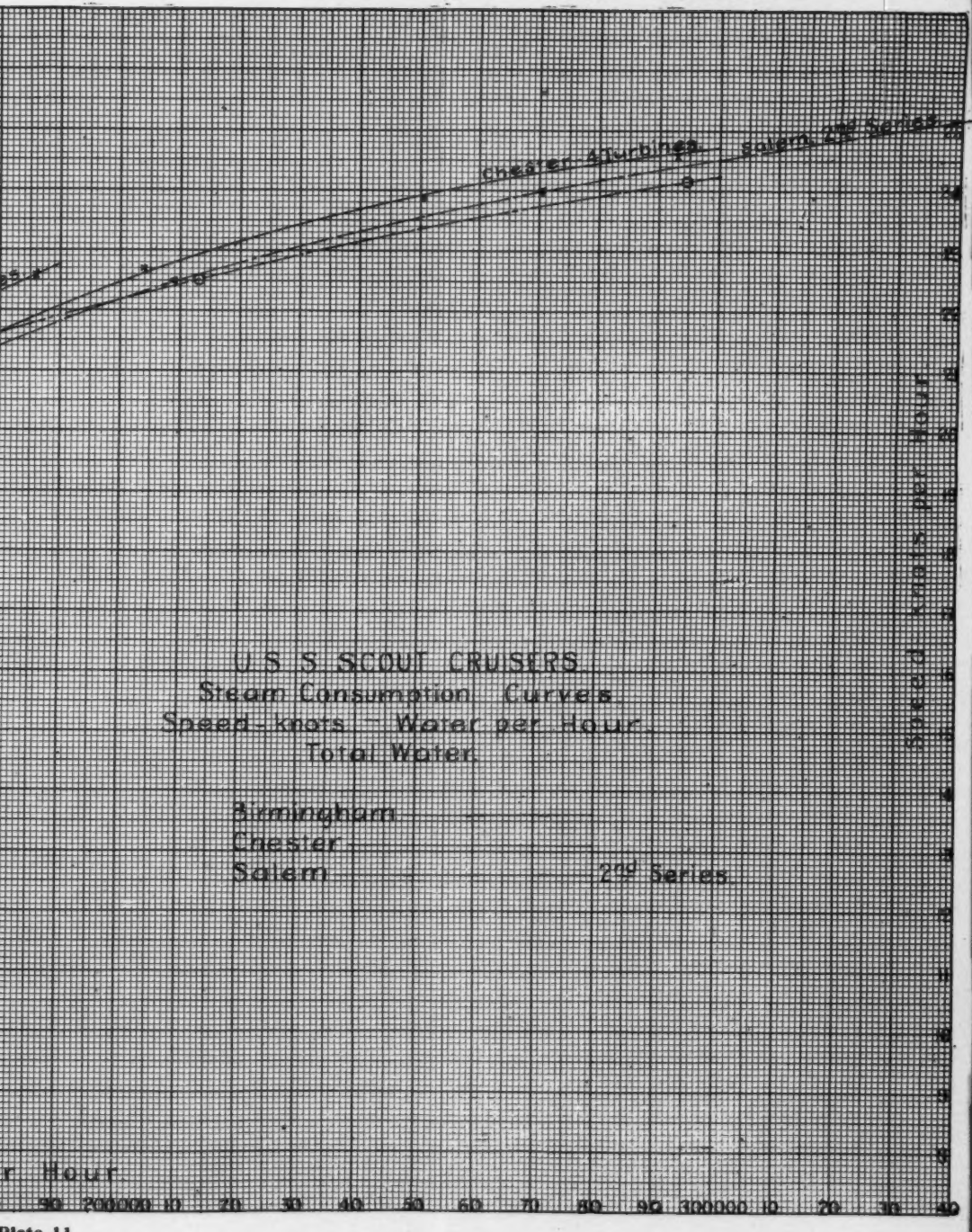
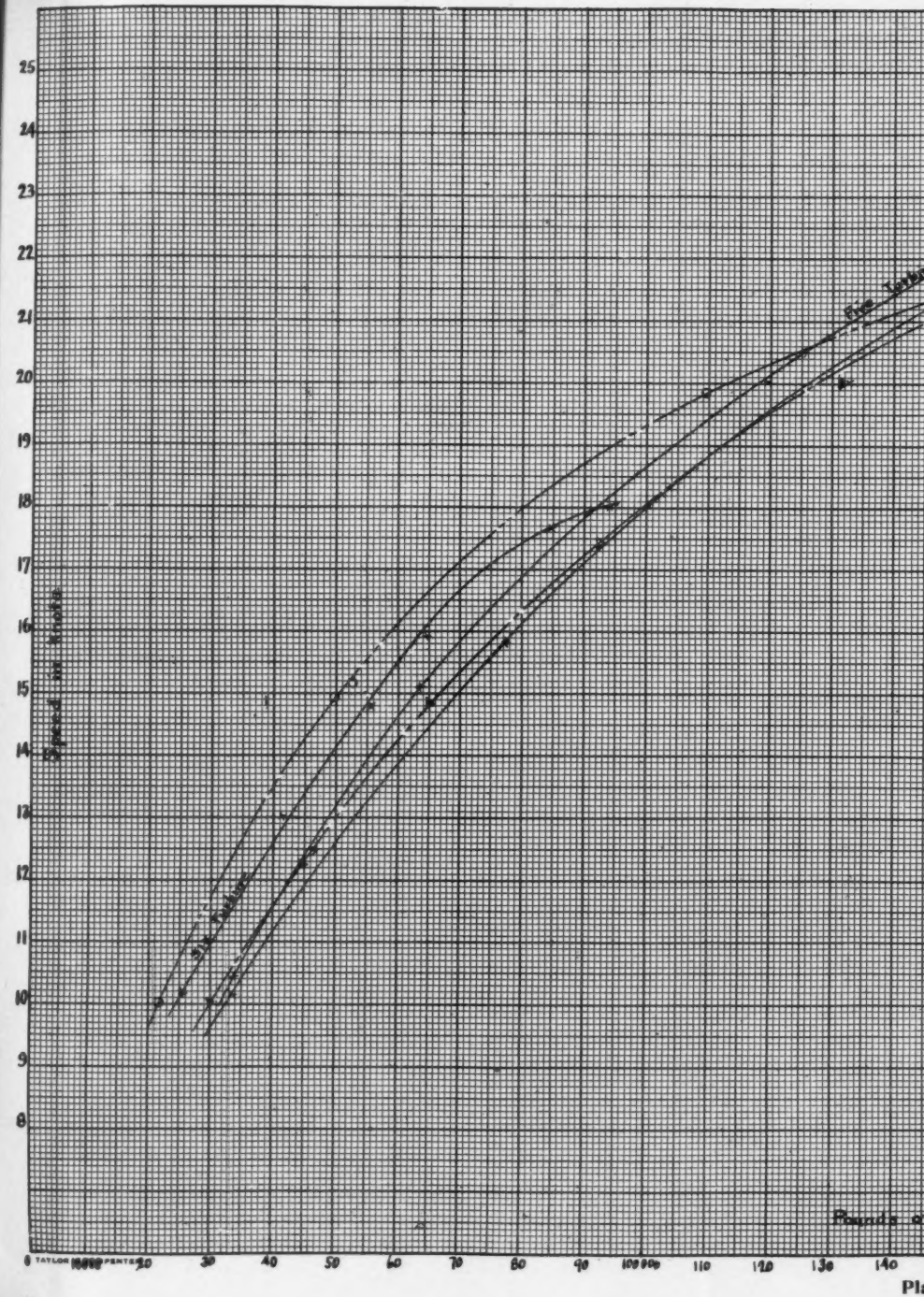

Chester Afturbines
Salem 2nd Series
U S S SCOUT CRUISERS
Steam Consumption Curves
Speed - knots - Water per Hour.
Total Water
Birmingham
Chester
Salem 2nd Series
Speed in Knots per Hour
r Hour
30 200000 10 20 30 40 50 60 70 80 90 300000 10 20 30 40

Plate 11.


25
24
23
22
21
20
19
18
17
16
15
14
13
12
11
10
9
8
Pounds per square inch
Pounds
0
TAYLOR
PENTA
20
30
40
50
60
70
80
90
110
120
130
140

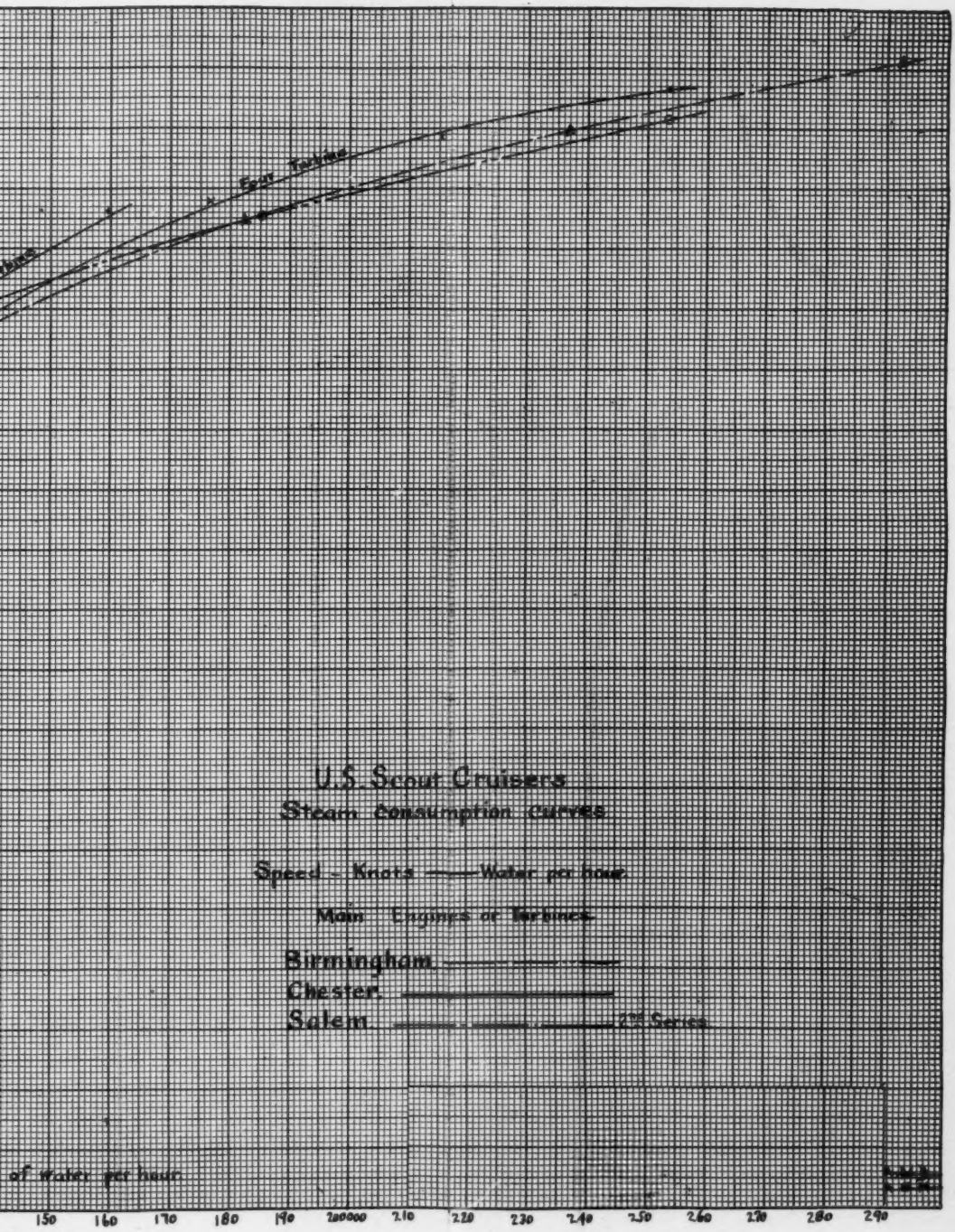
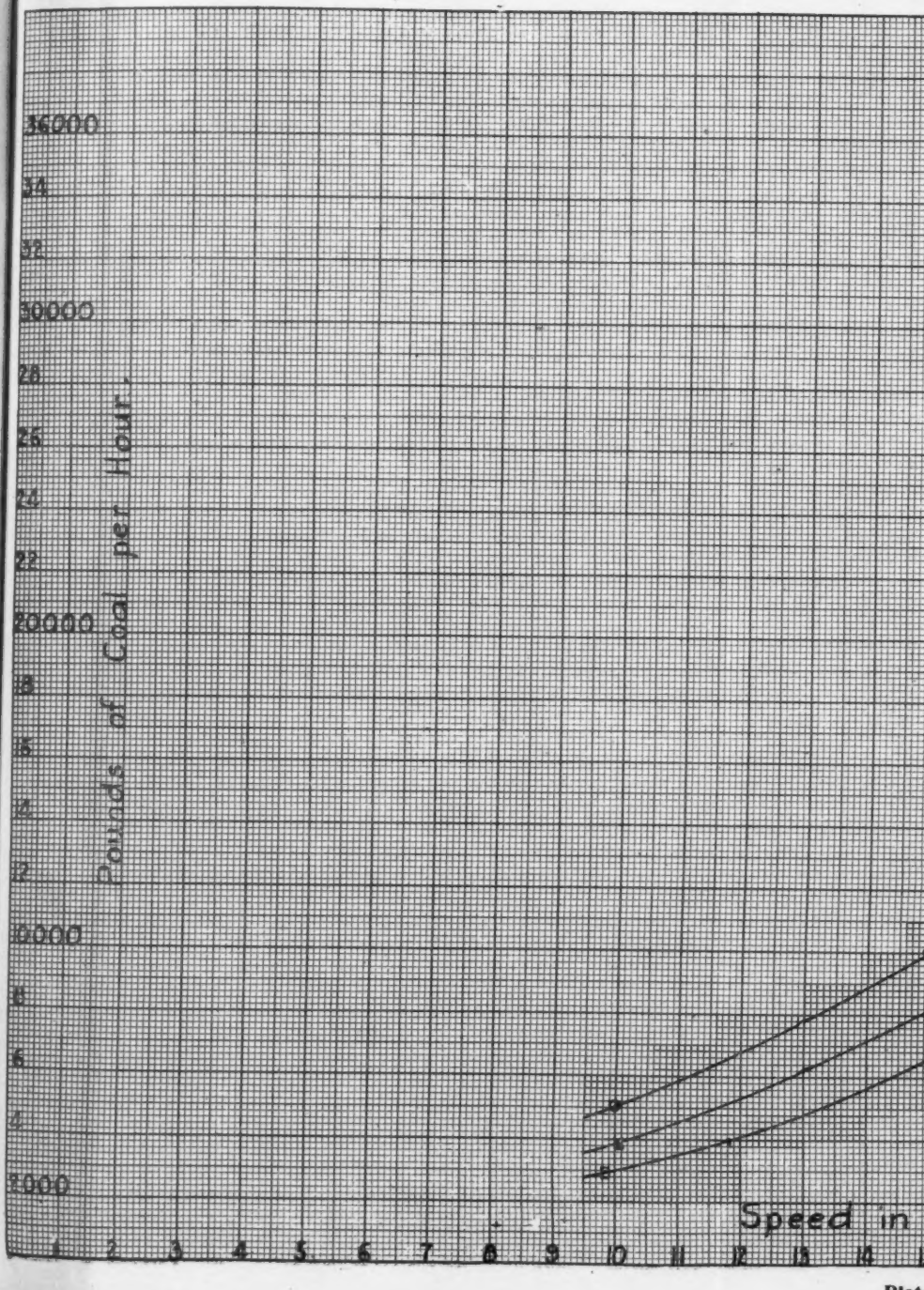
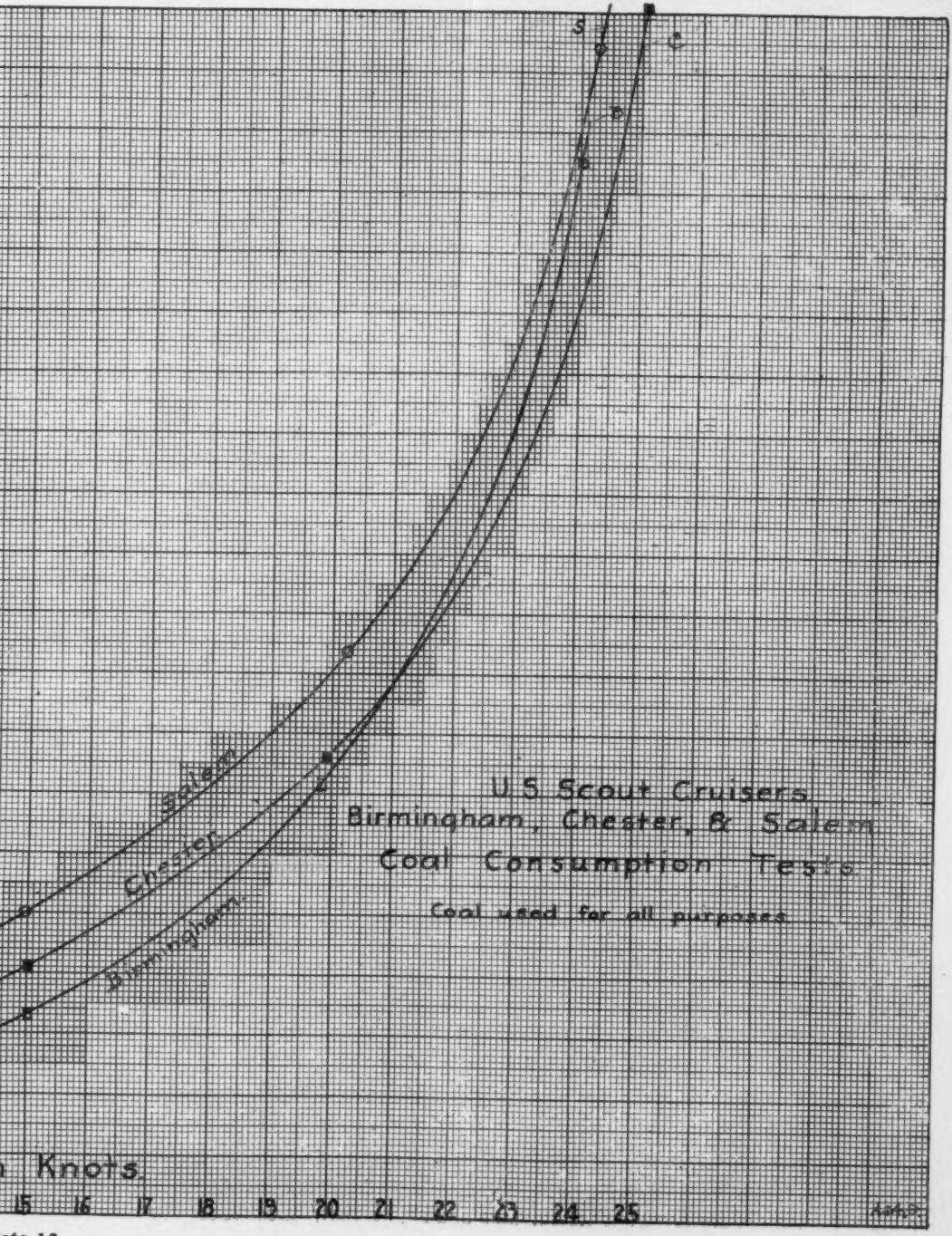

Fast Torpedo
U.S. Scout Cruisers
Steam Consumption Curves.
Speed - Knots ——Water per hour.
Main Engines or Turbines.
Birmingham.
Chester.
Salem. 2nd Series.
of water per hour.
150 160 170 180 190 200000 210 220 230 240 250 260 270 280 290

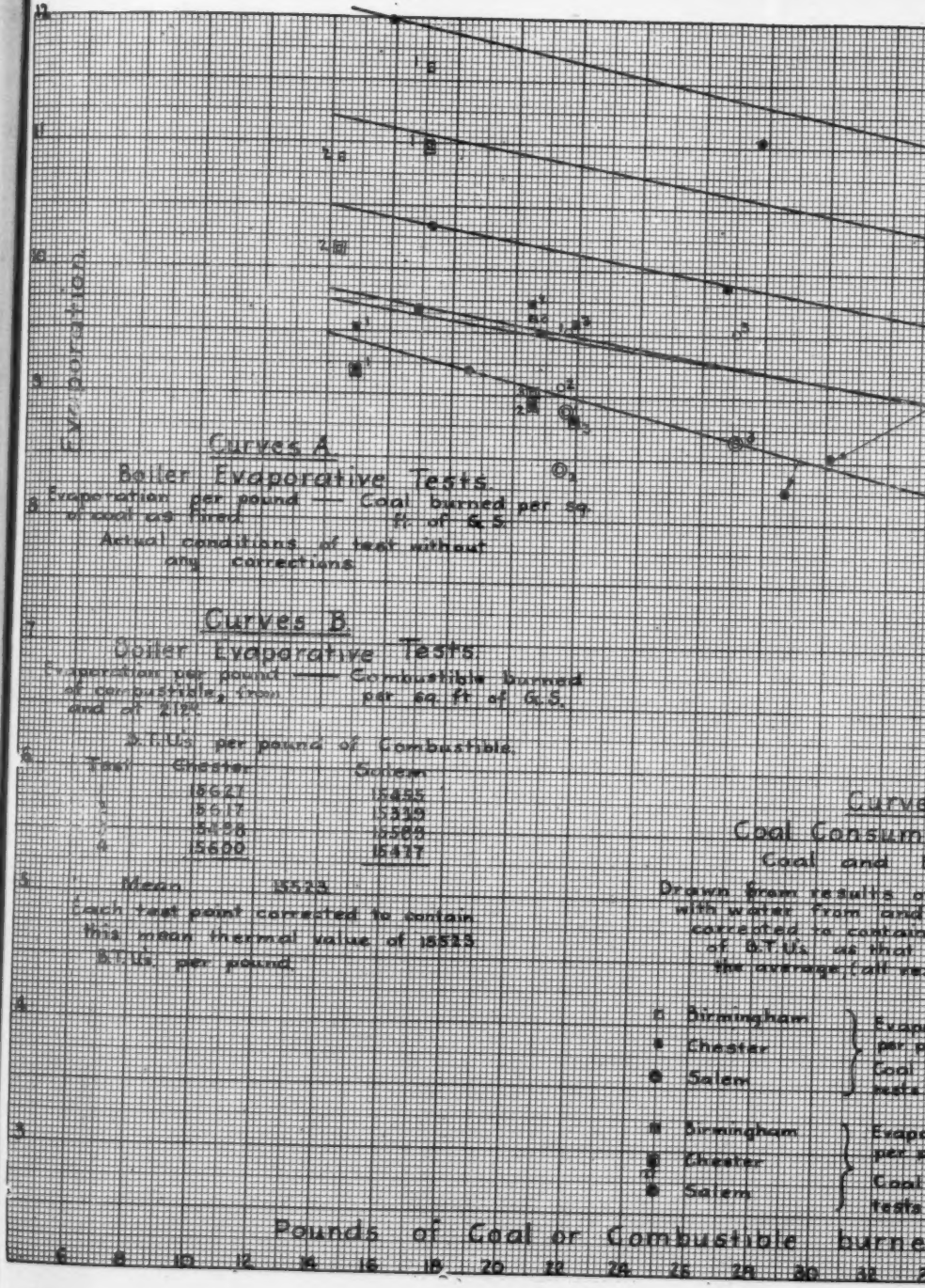
Plate 12.


36000
34
32
30000
28
26
24
22
20000
18
16
14
12
10000
8
6
4
2
1000
Pounds of Coal per Hour.
1 2 3 4 5 6 7 8 9 10 11 12 13 14
Speed in

Plat


U.S. Scout Cruisers
Birmingham, Chester, & Salem
Coal Consumption Tests
Coal used for all purposes
Salem
Chester
Birmingham
Knots.
15
16
17
18
19
20
21
22
23
24
25

ate 13.


Evaporation
Curves A.
Boiler Evaporative Tests.
Evaporation per pound — Coal burned per sq. ft of G.S.
of coal as fired.
Actual conditions of test without any corrections.
Curves B.
Boiler Evaporative Tests.
Evaporation per pound — Combustible burned per sq. ft of G.S.
of combustible, from and at 212°.
B.T.Us. per pound of Combustible.
Test Chester. Salem.
1 15627 15495
2 15617 15339
3 15498 15569
4 15600 15427
Mean 15535.
Each test point corrected to contain this mean thermal value of 15535 B.T.Us. per pound.
Curve
Coal Consum
Coal and
Drawn from results o
with water from and
corrected to contain
of B.T.Us. as that
the average Coal te
Birmingham
Chester
Salem
Evap
per p
Coal
tests
Birmingham
Chester
Salem
Evap
per
Coal
tests
Pounds of Coal or Combustible burne
6 8 10 12 14 16 18 20 22 24 26 28 30 32

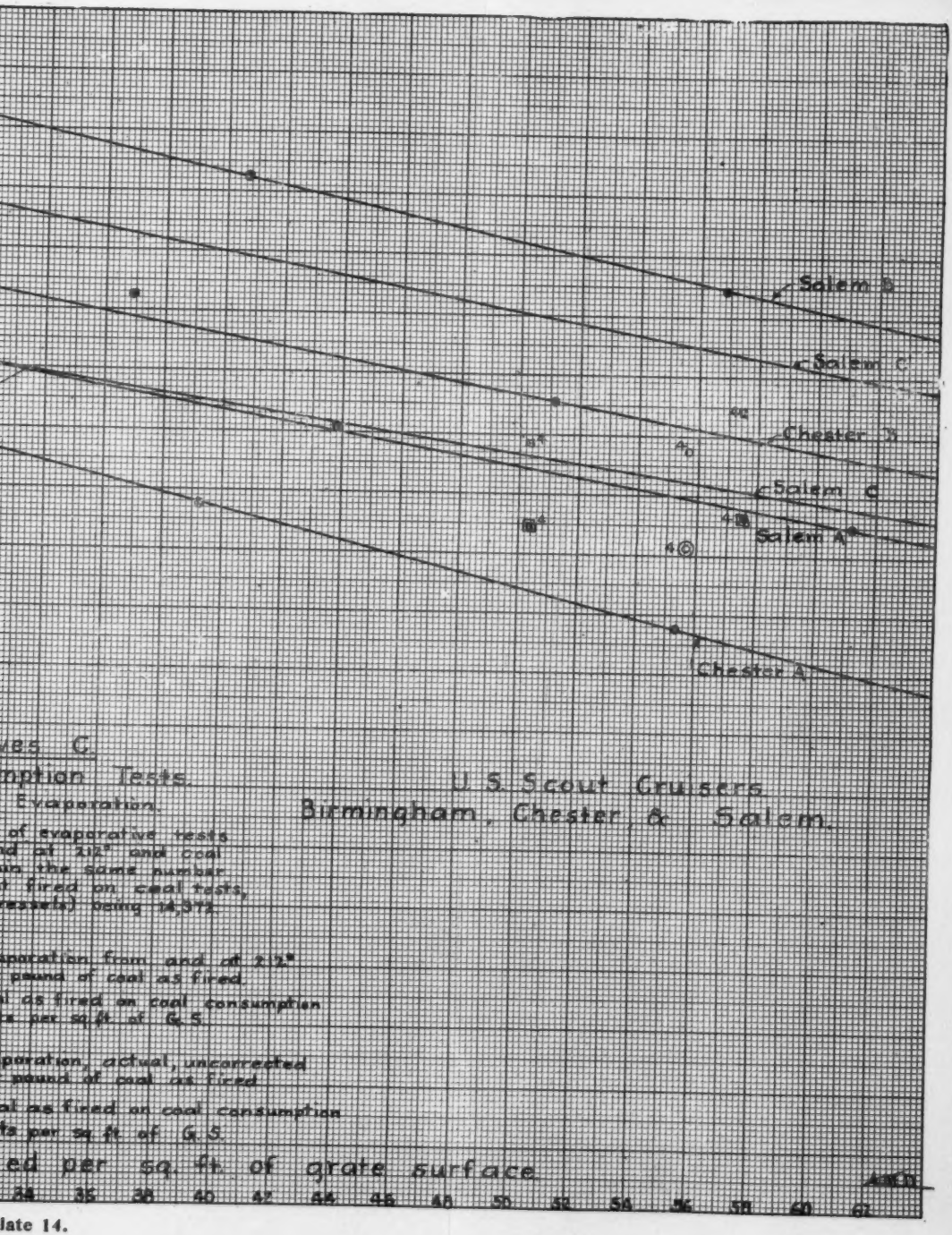

Salem B
Salem C
Chester B
Salem C
Salem A
Chester A
ves C.
mption Tests.
U S. Scout Cruisers
Birmingham, Chester, & Salem.
Evaporation.
of evaporative tests
d at 212° and coal
un the same number
fired on coal tests,
essels) being 14,97½.
aporation from and at 212°
pound of coal as fired.
l as fired on coal consumption
s per sq ft of G.S.
poration, actual, uncorrected
pound of coal as fired
al as fired on coal consumption
ts per sq ft of G.S.
ed per sq. ft. of grate surface.
28 36 38 40 42 44 46 48 50 52 54 56 58 60 62

Plate 14.

BACKING TESTS OF SCOUT CRUISERS.

Compiled from Reports.

In order to obtain reliable data on the relative backing power of vessels equipped with reciprocating engines—Parsons turbines and Curtis turbines—tests were ordered by the Navy Department to be made with the three scout cruisers *Birmingham, Chester* and *Salem.* The conditions prescribed were as follows: All vessels to be under similar conditions as to trim and displacement, and the trials to be conducted at approximately 10 knots, 16 knots, 22 knots, and the maximum speed obtainable for all vessels. The vessels were to steam at the prescribed speeds, with only the number of boilers under steam that would be required were the speed to be maintained for an indefinite period. All engines to be reversed at the maximum power available and to continue backing, until the vessel was dead in the water, at the maximum speed possible without reducing the steam pressure in the boilers to such a degree as to cause priming. The following data was ordered taken:

(a) Displacement.

(b) Trim.

(c) Condition of bottom and time out of dock.

(d) Number of boilers connected.

(e) Elapsed time from signal to reverse until engines moved astern.

(f) Elapsed time from signal to reverse until vessel was dead in the water.

(g) Distance headreached during the backing interval.

(h) Steam pressure and vacuum at beginning and end of backing interval.

The tests were carried out as ordered. The maximum speed which could be obtained without difficulty, under stand-

ard conditions, was estimated to be 24 knots, and this was the speed at which the last series of runs were made. At the time of the trials there were calms and light airs to gentle breezes with a ground swell making the vessels, while running, roll from 3 to 10 degrees and pitch from 1 to 2 degrees. The distance headreached for each vessel was calculated from observations taken from the other vessels, at the instant of giving signal for reversing and when ship became stationary. The observing vessels lay at anchor, and the runs were made on a course such as to ensure accuracy of observations.

General descriptions of the machinery of these vessels have been given in the JOURNAL as follows: *Birmingham,* Volume XX, No. 3, page 706; *Chester,* Volume XX, No. 2, page 349; *Salem,* Volume XX, No. 4, page 976.

The propeller data for the three vessels is as follows:

Vessel.	No.	Diameter.	Pitch.	Developed area.	Projected area.
		feet, ins.	*feet, ins.*	*square feet.*	*square feet.*
Birmingham	2	12-6	15-3	49.4	40.8
Chester	4*	6-0	6-0	19.0	17.02
Salem	2	9-6	8-8½	43.7	36.8

*Two only used in backing.

The results obtained are given in the following table. It will be noted from the column giving time until ship stopped that the best results were obtained with the *Birmingham,* fitted with reciprocating engines, the *Salem,* with Curtis turbines coming second, and the *Chester,* with Parsons turbines, coming last. If we deduct the time required to reverse engines, thus obtaining the time engines were backing, it will be noted that the *Salem,* with Curtis turbines, stands first, *Birmingham,* with reciprocating engines, second, and *Chester,* with Parsons turbines, as before, last. The average time required to reverse engines on the *Salem* was 43.67 seconds, on the *Chester* it was 10.52 seconds and on the *Birmingham* it was 5.54 seconds. The considerably longer time on the *Salem* was due to the type of maneuvering valves fitted, which, in future designs will be improved.

COMPARATIVE BACKING DATA.

.....	Revolutions per minute, going ahead.	Corresponding speed, in knots.	Time required to reverse engines.	Time engines were backing.	Total time until ship stopped.	Total revolutions while backing.	Revolutions per minute, backing.	Head reach of ship, in lengths.	Steam press. per gauge, boilers.		Vacuum in condenser.		Mean draught, feet and inches.	Displacement, tons.	Number of boilers in use.
									Before backing.	After backing.	Before backing.	After backing.			
1ST SERIES, 10 knots :			*s.*	*m. s.*	*m. s.*										
Birmingham, mean of 3 runs...	80.0	10.0	4.0	1:06.3	1:10.3	109.0	98.6	1.5	235.0	240.0	27.2	26.7	19–7½	4,855.5	6
Chester, mean of 4 runs.....	193.2	9.85	5.0	1:47.7	1:52.7	492.3	274.2	2.2	230.0	115.7	28.7	26.7	19–8	4,950.0	6
Salem, mean of 3 runs.....	142.0	10.0	35.2	0:45.2	1:20.4	136.2	180.8	1.8	270.8	181.7	29.02	27.29	19–8	4,877.0	6
2D SERIES, 16 knots :															
Birmingham, mean of 3 runs...	130.0	16.07	4.67	1:16.53	1:21.2	139.2	109.1	2.6	250.0	230.0	27.4	27.3	19·7	4,851.0	8
Chester, mean of 3 runs.....	309.4	15.79	9.1	1:56.9	2:06.0	528.7	271.4	4.3	238.3	143.3	28.38	23.37	19–6½	4,785.0	8
Salem, mean of 3 runs.....	230.0	16.0	42.0	0:45.3	1:27.3	156.4	207.1	3.0	280.8	206.7	29.0	27.52	19-7	4,846.0	8
3D SERIES, 22 knots :															
Birmingham, mean of 2 runs...	184.0	21.75	6.5	0:56.5	1:03.0	140.0	148.7	3.0	250.0	250.0	25.25	25.11	19–6½	4,832.5	12
Chester, mean of 2 runs.....	446.2	22.08	13.6	1:54.9	2:08.5	542.5	283.3	5.4	252.0	245.0	28.47	27.85	19–4	4,785.0	12
Salem, mean of 2 runs.....	332.0	22.09	47.5	0:49.5	1:37.0	174.3	211.25	4.2	272.5	242.0	27.75	26.83	19–7	4,846.0	12
4TH SERIES, 24 knots :															
Birmingham, 1 run.....	205.0	23.4	7.0	1:02.0	1:09.0	156.5	151.4	3.2	250.0	250.0	24.5	24.5	19–6⅜	4,832.5	12
Chester, 1 run.....	500.8	24.25	14.4	2:16.6	2:31.0	684.5	300.7	6.5	255.0	250.0	28.25	26.0	19–4	4,785.0	12
Salem, 1 run.....	369.0	24.0	50.0	0.50.5	1:40.5*	170.4	202.5	4.7	275.0	257.0	27.75	26.0	19–7	4,846.0	12

*Starboard engine, total, 214.6; R.p.m., 255; port engine, total, 126.2; R.p.m , 150.

It will be noted that the Curtis turbine had ample backing power, but the steam required was, when compared with the other vessels, abnormally large, as is evidenced by the drop in pressure. The turbine arrangement on the *Salem* is such that on the port turbine the unbalanced steam pressure is with the thrust from the propeller, while on the starboard shaft the steam pressure is against the thrust of the propeller. In the port turbine great care must be taken in backing to adjust the clearance, in order to avoid rubbing the blades, while in the starboard turbine such care is not required. For this reason the revolutions of the starboard turbine in backing was, in all cases, greater than in the port turbine, and in the fourth series this difference, as noted in the table, was so great as to give a result not strictly comparable with the results in the other series. In connection with the results obtained on the *Chester,* it should be noted that backing turbines are fitted only on the inboard shafts, two only of the four propellers being available for backing. In later installations of the Parsons turbines on battleships backing turbines are fitted on all shafts.

A METHOD OF ESTIMATING THE COAL CONSUMPTION OF VESSELS AT VARIOUS SPEEDS.

By Lieutenant Ormond L. Cox, U. S. N., Member.

Now that economy has become of such great importance in naval administration the subject of coal consumption is most vital not only to engineer officers but to every officer concerned in the direction of the movements of the fleet.

Various articles have appeared in this journal bearing on the question of coal consumption (see "Economic Marine Propulsion," John Lowe, Vol. III, p. 218; "Coal Endurance and Machinery of New Cruisers," etc., Vol. IV, p. 637, and "Economic Speed and Power of War Vessels," etc., I. N. Hollis, Vol. V, p. 66; "The Steaming Radius of United States Naval Vessels," B. C. Bryan, Vol. XIII, p. 50; "Speed and Coal-Consumption Curves," D. S. Mahony, Vol. XV, p. 366; "A Method of Calculating the Coal Endurance and Steaming Radius of War Vessels under Various Conditions," B. C. Bryan, Vol. XVII, p. 696.) In practically all cases where a mathematical investigation of the subject has been attempted, curves have been deduced of the form $y=k+CR$, where y is the total coal consumption in tons per day, k and C are constants, and R the revolutions per minute or speed in knots per hour.

The values of n, so deduced, lie between 2 and 3, and, up to a speed of about twelve knots, give a very satisfactory curve, but at greater speeds than twelve knots the coal consumption increases in a higher ratio than the cube of the speed. My purpose is to show that an equation may be deduced that will contain all possible powers of the speed. The general equation will be of the form $k = A + BV + CV^2 + DV^3 +$ etc.; but this will not be easy to handle and will necessitate the use of accurate data obtained at very low speeds in order to give the form of the curve at these speeds.

The effective horsepower of a ship is given by $\frac{RV}{33{,}000}$, where R is the resistance of the ship in pounds, and V the speed of the ship in feet per minute. Since R varies approximately as V^2, we have the effective power varying approximately as V^3. The coal consumption, being proportional to the power delivered, also varies approximately as V^3. This gives us one term of our equation, viz: CV^3, where C is a constant, and V the speed in knots per hour.

To obtain the remainder of the equation we assume that there is a definite "limiting speed" for the vessel, beyond which it is impossible to drive the ship as equipped, even though the coal consumption be increased indefinitely. This assumption naturally leads to a term the denominator of which is of the form $B - V$, where B is a constant and V the speed as before. When V is equal to B the denominator becomes zero and the term infinite.

We can now write the general equation for the coal-consumption curve as follows:

$k = \frac{A}{B - V} + CV^3$, where k is the coal consumption in tons per day, V the speed of the ship in knots per hour; A, B and C are constants to be determined from actual runs.

It is to be noted that this equation contains all possible powers of V, for by expanding the factor $\frac{A}{B - V}$ we get $\frac{A}{B - V} = \frac{A}{B} + \frac{AV}{B^2} + \frac{AV^2}{B^3} + \frac{AV^3}{B^4} + \text{etc.}$

In this expression it will be noted that at low speeds the coefficient decreases rapidly as the power of V increases, and soon approximates to zero.

The factor $\frac{A}{B}$ is the coal consumption at zero speed, and is obtained by making V equal zero in the equation for coal consumption.

To facilitate solving the equation put it in the form: $Bk - Vk = A + BCV^3 - CV^4$. Take a sheet of cross-sec-

tion paper and plot a curve of actual consumption from the data given. The data should contain the actual observed coal consumption at three or four different speeds, one of which should be near the maximum speed and one at least as low as half speed. From the curve thus plotted obtain the coal consumption at four speeds at intervals of three or four knots, and by substituting in the above equation the constants A, B and C are obtained.

The actual coal consumption is seldom obtained for an even-knot speed, but usually for fractions, thus involving considerable labor in obtaining the third and fourth powers of the speed; but by plotting the curve we are able to obtain the consumption for the even knot.

We now have the general equation for the coal-consumption curve, but coal consumption is a variable quantity even under apparently the same conditions. In constructing the curve it is necessary to know just what machinery is in use. This should be all machinery necessary to propulsion, including the dynamos, evaporators, etc.

The coal consumption of the dynamos should be that which is necessary for daily use, lighting the ship, running auxiliaries, etc., but should not include any temporary extra load, as for turret training or running deck winches.

The coal consumption of the evaporators should be that necessary for make-up feed and the daily average for ship's use.

Considering these factors, we see that practically all of them are variable. For example, the amount of make-up feed depends on the condition of all joints, stuffing boxes, etc. In the same way, the general condition of all the machinery affects the coal consumption. The personal element in firing is another large variable.

Having once obtained a curve when everything is in good condition, every senior engineer officer should strive to keep the coal consumption down to the curve, or even below it.

To modify the curve for any constant increase in coal consumption due to extra load on the dynamos, increase in

amount of distilled water, etc., all that is necessary is to lower the base line of the curve by the amount of extra coal used for this purpose.

The coal necessary for propulsion depends on the revolutions maintained.

As the ship's bottom becomes foul the speed drops for the same coal consumption, but the revolutions fall off in a much smaller ratio. In figuring on economy of coal consumption it will, therefore, be found of advantage to estimate on "economical revolutions" instead of economical speed.

In obtaining the coal consumption from the curve under various conditions, we should not use the speeds themselves, but the speed that the same number of revolutions give on a speed-revolution curve constructed simultaneously with the coal-consumption curve.

The most accurate data on hand on coal consumption is that obtained on the trials of the scout cruisers *Birmingham*, *Chester* and *Salem*, though this is not absolutely accurate, since all coal was measured by marks in the bunkers instead of being weighed. The accuracy of the results on the *Salem* was impaired by the condition of her turbines at the time of the trials.

These ships were run under service conditions, so that the curves plotted from the data as given will be correct.

The data for the *Birmingham* and *Chester*, given in Table I, is taken from the Report of the Board on Comparative Trials of the Scout Cruisers *Birmingham*, *Salem* and *Chester*.

Below is given the solution for the curve in the case of the *Birmingham*. From the data given a curve of actual consumption is constructed, from which we obtain the following data:

Speed of ship, in knots per hour.	Coal, tons per day.
24	375.72
20	157.25
16	84.50
12	45.00

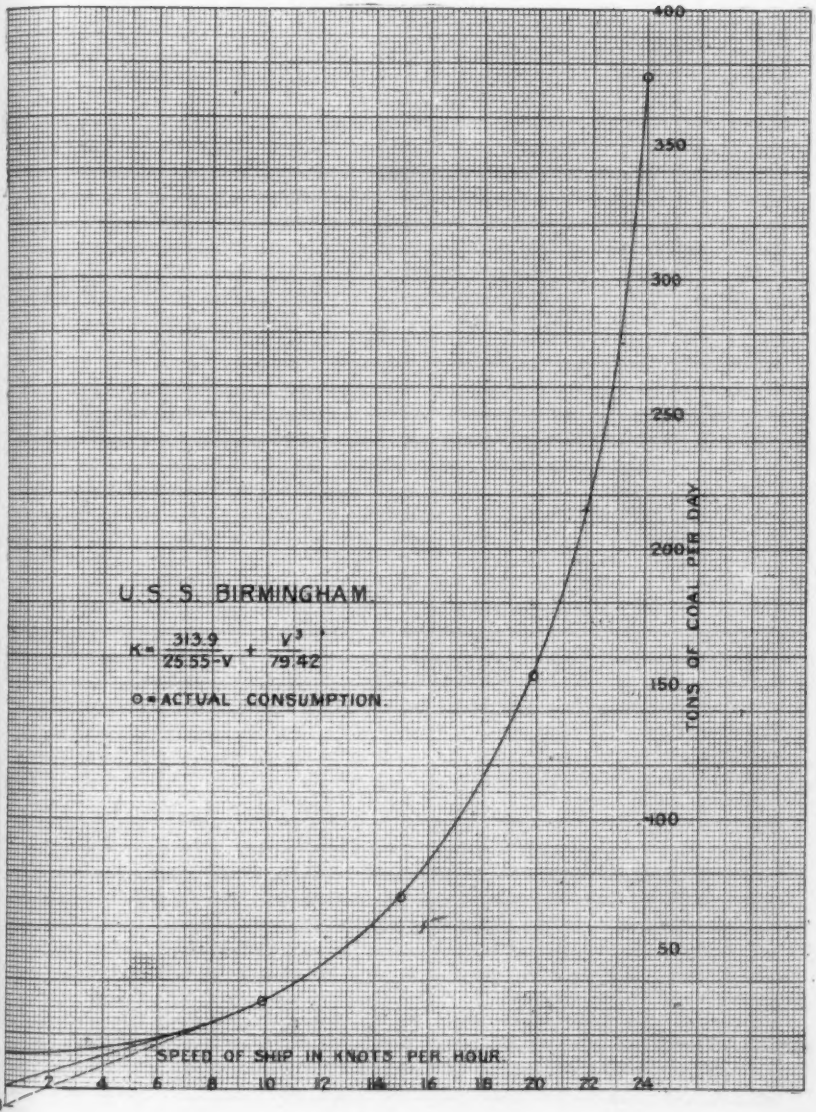

U.S.S. BIRMINGHAM.
K = 313.9 / (25.55 - V) + V³ / 79.42
○ = ACTUAL CONSUMPTION.
TONS OF COAL PER DAY.
400
350
300
250
200
150
100
50
SPEED OF SHIP IN KNOTS PER HOUR.
2
4
6
8
10
12
14
16
18
20
22
24

Plate 1.

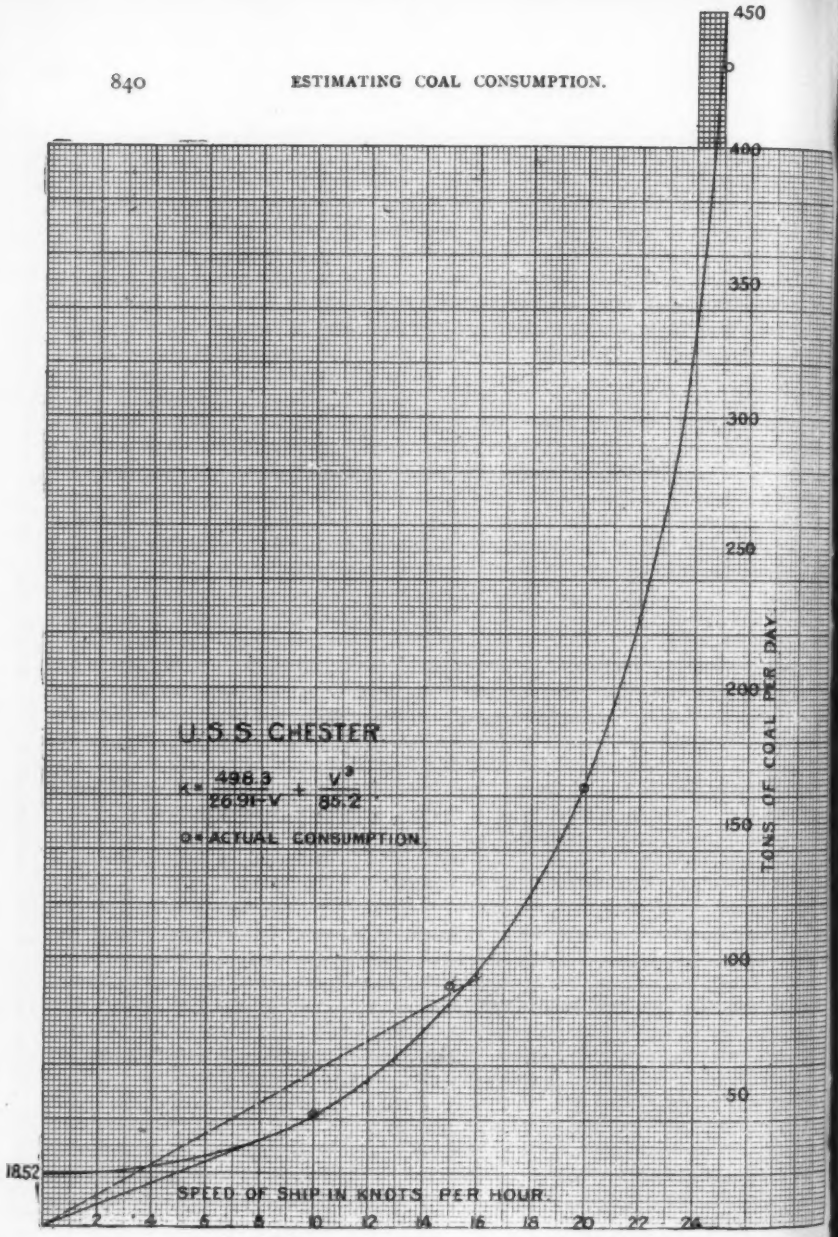

450
400
350
300
250
200
150
100
50
TONS OF COAL PER DAY.
U.S.S. CHESTER
K = 498.3 / 26.9 F·V + V³ / 85.2
o = ACTUAL CONSUMPTION
18.52
SPEED OF SHIP IN KNOTS PER HOUR.
2
4
6
8
10
12
14
16
18
20
22
24

Plate 2.

Substituting these values in the general equation, $Bk - Vk = A + BCV^3 - CV^4$, we obtain four equations, as follows:

$$375.72\,B - 9{,}017.28 = A + 13{,}824\,BC - 331{,}776\,C \quad . \quad (1)$$
$$157.25\,B - 3{,}145.00 = A + 8{,}000\,BC - 160{,}000\,C \quad . \quad (2)$$
$$84.50\,B - 1{,}352.00 = A + 4{,}096\,BC - 65{,}536\,C \quad . \quad (3)$$
$$45.00\,B - 540.00 = A + 1{,}728\,BC - 20{,}736\,C \quad . \quad (4)$$

Subtracting (4) from (3), (3) from (2) and (2) from (1), we get:

$$39.50\,B - 812.00 = 2{,}368\,BC - 44{,}800\,C \quad . \quad (5)$$
$$72.75\,B - 1{,}793.00 = 3{,}904\,BC - 94{,}464\,C \quad . \quad (6)$$
$$218.47\,B - 5{,}872.28 = 5{,}824\,BC - 171{,}776\,C \quad . \quad (7)$$

To eliminate the factor BC use the multipliers from Table II. Equation (5) is to be multiplied by (a) and equation (6) by (b).

$$218.47\,B - 5{,}872.28 = 5{,}824\,BC - 171{,}776\,C \quad . \quad (7)$$
$$108.53\,B - 2{,}674.80 = 5{,}824\,BC - 140{,}921\,C \quad . \quad (6')$$
$$97.15\,B - 1{,}997.08 = 5{,}824\,BC - 110{,}184\,C \quad . \quad (5')$$

Subtracting (5′) from (7), (6′) from (7) and (5′) from (6′), we get:

$$121.32\,B - 3{,}875.20 = (-)\ 61{,}592\,C \quad . \quad (8)$$
$$109.94\,B - 3{,}197.48 = (-)\ 30{,}855\,C \quad . \quad (9)$$
$$11.38\,B - 677.72 = (-)\ 30{,}737\,C \quad . \quad (10)$$

To eliminate C multiply equation (9) by multiplier (c) and equation (10) by (d), which gives:

$$121.32\,B - 3{,}875.20 = (-)\ 61{,}592\,C \quad . \quad (8)$$
$$219.47\,B - 6{,}383.04 = (-)\ 61{,}592\,C \quad . \quad (9')$$
$$22.80\,B - 1{,}358.00 = (-)\ 61{,}592\,C \quad (10')$$

From which we get:

$$\left.\begin{aligned} 99.15\,B - 2{,}507.84 &= 0 \\ 98.52\,B - 2{,}517.20 &= 0 \\ 196.67\,B - 5{,}025.04 &= 0 \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad B = 25.55$$

Substitute the value of B in equations (8), (9) and (10) and get the value of $C = \frac{1}{79.42}$. Then using the original equation and substituting the values of B and C as found we get A. $k = \frac{A}{25.55 - V} + \frac{V^3}{79.42}$. We get four values for A, as follows: 312.5, 313.7, 314.4 and 315. Average, 313.9, which gives the equation $k = \frac{313.9}{25.55 - V} + \frac{V^3}{79.42}$. Putting $V = 0$, we find $k = 12.28$ = the coal expenditure at zero speed, or the coal for auxiliaries independent of the speed.

In the same way we obtain the curve for the *Chester* from the following data:

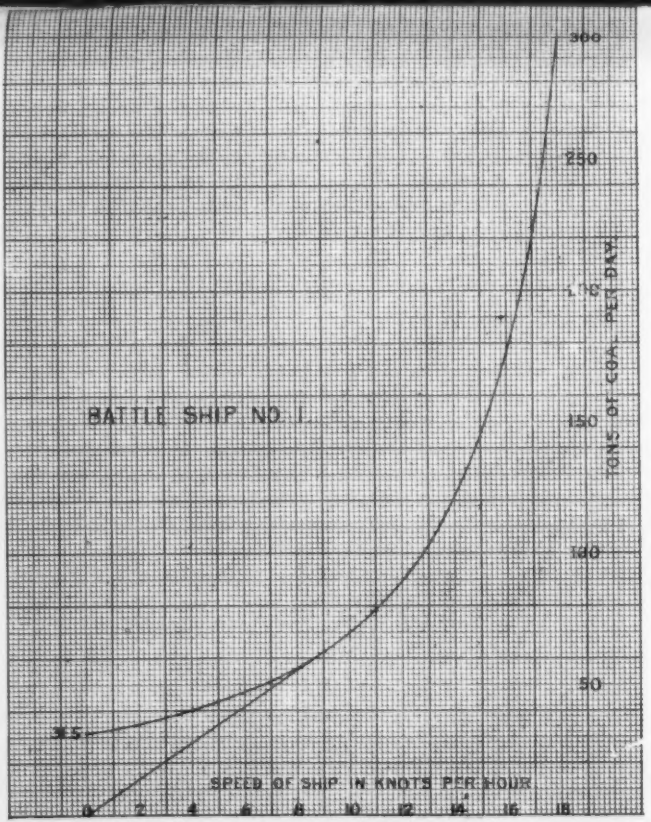
Speed, in knots per hour.	Coal, tons per day.
24	332
20	166
16	94
12	54

which gives the equation $k = \frac{498.3}{26.91 - V} + \frac{V^3}{85.2}$.

The curve for the *Birmingham* is plotted on Plate 1 and that for the *Chester* on Plate 2.

The economical speed of a vessel is that speed for which the steaming radius is a maximum, and is given by the point of contact of the tangent drawn from the origin to the curve. This is only true when the ship's bottom is clean. When the ship's bottom is foul the economical speed is that speed which the revolutions necessary to drive the ship at the economical speed with clean bottom will drive the ship with foul bottom, *i. e.*, the economical revolutions.

In the case of the *Birmingham* the economical speed is 7 knots. The change of economical speed due to a change in the amount of coal necessary for auxiliary purposes is shown on the curve for the *Birmingham*. An increase of eight tons of coal per day for auxiliary purposes raises the economical speed from 7 knots to $8\frac{1}{2}$ knots.


BATTLE SHIP NO. 1.
300
250
200
150
100
50
TONS OF COAL PER DAY.
35
SPEED OF SHIP IN KNOTS PER HOUR.
0 2 4 6 8 10 12 14 16 18

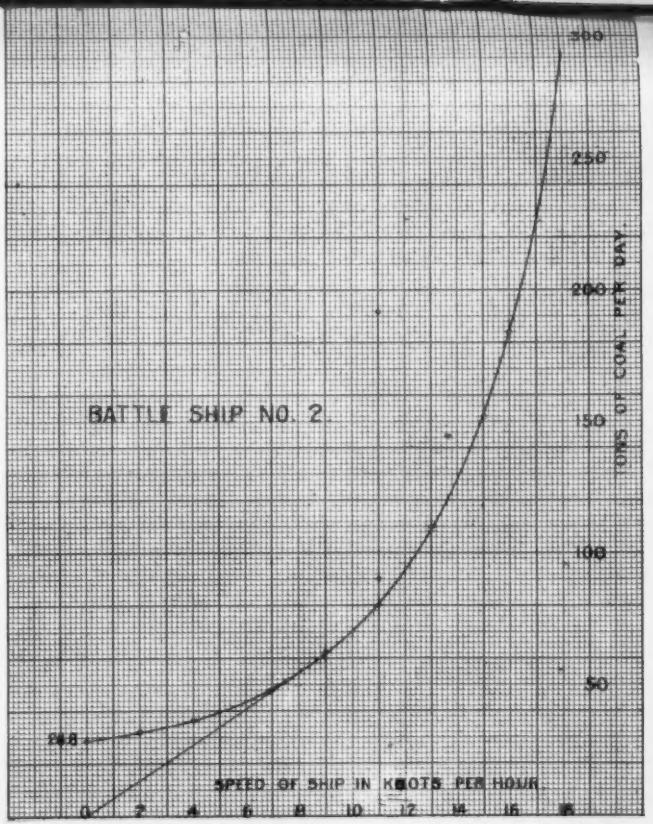
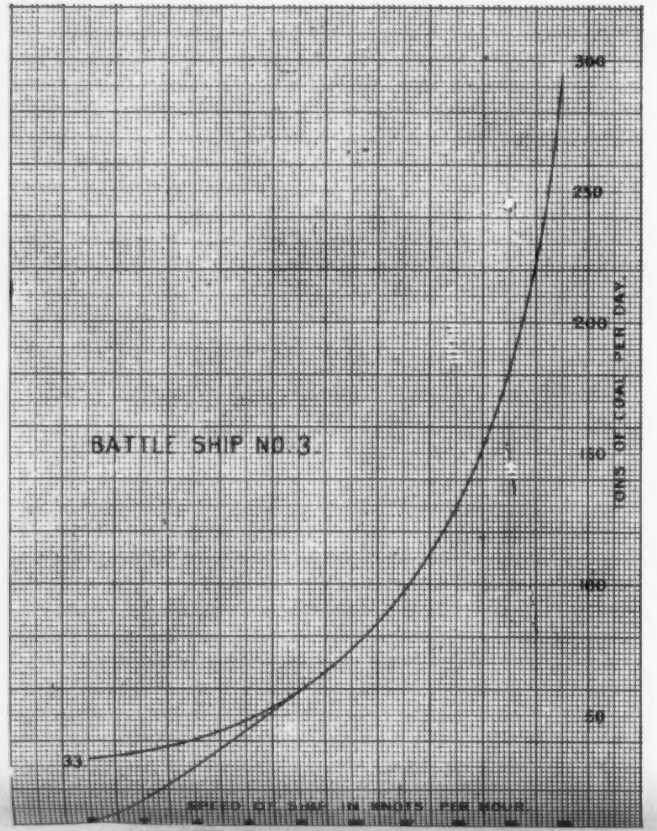
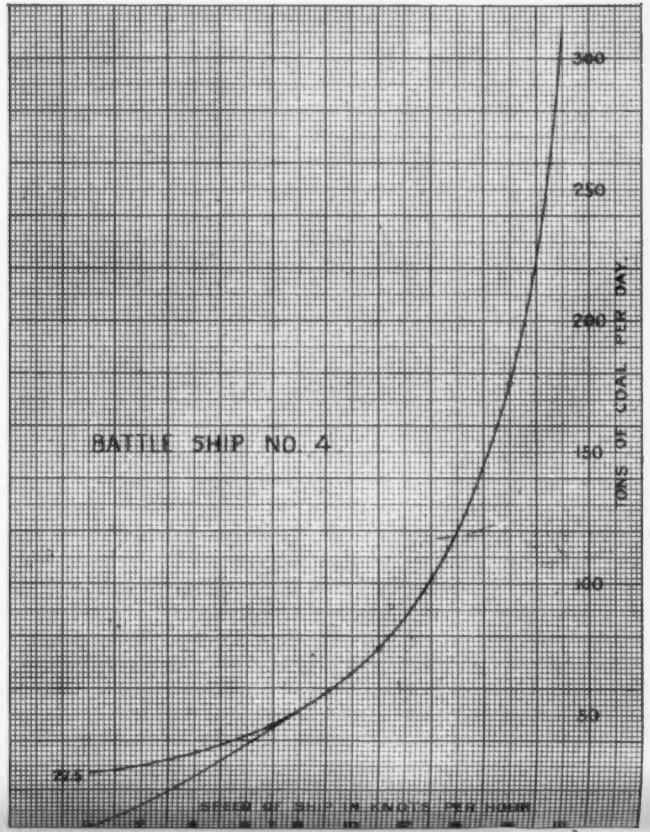

BATTLE SHIP NO. 2.
300
250
200
150
100
50
TONS OF COAL PER DAY.
28.8
SPEED OF SHIP IN KNOTS PER HOUR.
0 2 4 6 8 10 12 14 16 18

Plate 3.


BATTLE SHIP NO. 3.
300
250
200
150
100
50
33
TONS OF COAL PER DAY.
SPEED OF SHIP IN KNOTS PER HOUR.


BATTLE SHIP NO. 4
300
250
200
150
100
50
24.5
TONS OF COAL PER DAY.
SPEED OF SHIP IN KNOTS PER HOUR.

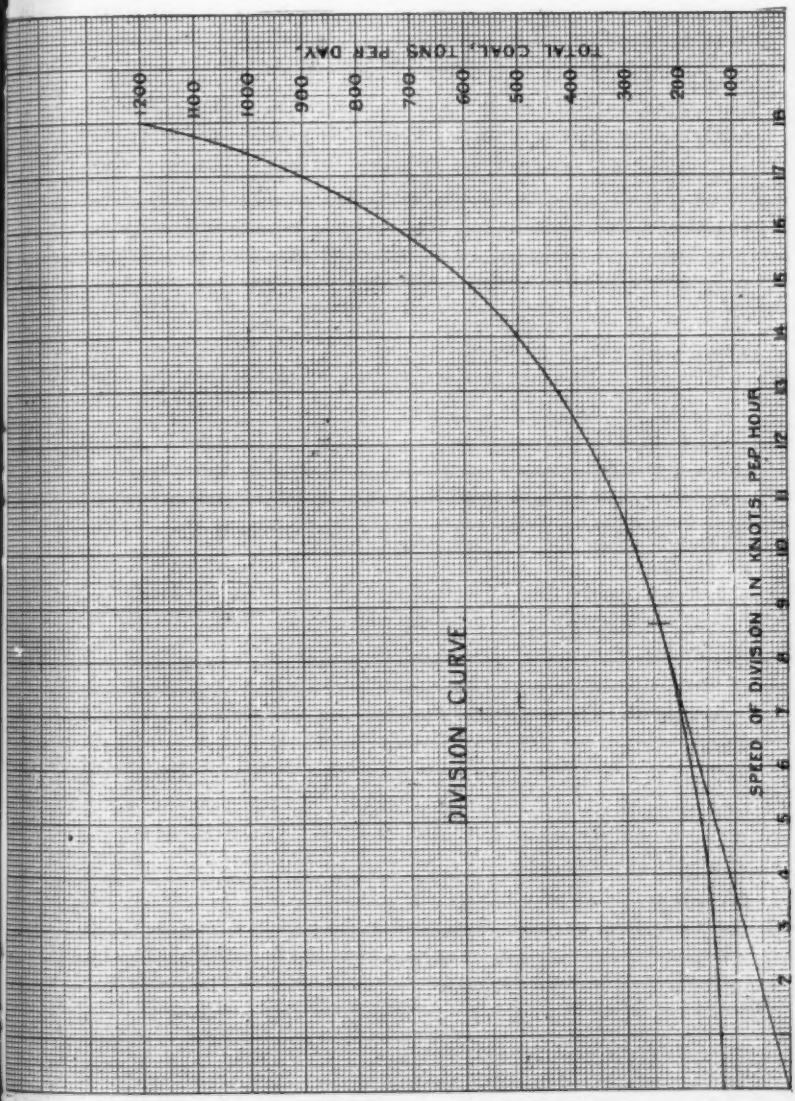

TOTAL COAL, TONS PER DAY.
1200
1100
1000
900
800
700
600
500
400
300
200
100
DIVISION CURVE.
SPEED OF DIVISION IN KNOTS PER HOUR.
2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18

Plate 5.

It should be noted that, for a range of three or four knots there is very little difference in economy.

Cruising Speeds.—For the *Chester* (Plate 2) the economical speed is seen to be nearly 8½ knots. Drawing any straight line from the origin, cutting the curve in two points will give two speeds of the same economy. That is, with a given amount of coal the ship would steam the same distance at either speed.

Under favorable conditions the ship should always be run at the higher of the two speeds, for a variation of one knot speed at the lower speeds corresponds to a variation of several knots at the higher speeds.

The dotted line on the *Chester's* curve shows that the same economy is obtained at 15.7 knots speed as at 4 knots.

The curves shown on Plates 3 and 4 are not curves of actual ships, but are assumed curves for four battleships, and Plate 5 shows these curves combined into a "division curve" by simply adding together the various coal consumptions at the given speeds.

For battleship No. 1 the economical speed is 8.9 knots.

For battleship No. 2 the economical speed is 8.0 knots.

For battleship No. 3 the economical speed is 8.5 knots.

For battleship No. 4 the economical speed is 8.5 knots.

While the economical speed for the division is about 8.6 knots.

This is not the economical speed of any vessel, but is the best speed for the division as a whole. If the Division Commander has the combined curve for his division he could tell at a glance what speed he should make for the best economy, or, rather, what revolutions should be made. By combining the "division curves" the Commander-in-Chief could obtain a "fleet curve," showing the same for the fleet as the division curve shows for the division.

In drawing the actual curve with a spline, having given only three or four points, it will be found that there is an infinite number of curves that may be drawn through the points, though they differ very little in general outline, and

the various coal consumptions may differ by a few tons at the different speeds. It is easy to determine whether the right values for coal consumption have been used, for, though the different values of B and C obtained from the equations are practically the same from each equation, the values of A will differ widely unless the proper values have been used. A little experience will soon show which coal consumption should be increased and which decreased, and by how much, in order to make the four values of A practically the same.

In making the multiplications the use of a Fuller's calculator is recommended. These are furnished nearly all ships, save much time and trouble, and are sufficiently accurate for the work required.

TABLE I.

	Speed, knots per hour.	Coal, tons per day.
Birmingham,	9.86	32.14
	15.00	71.21
	19.83	153.45
	24.00	375.72
Chester,	10.03	42.06
	14.98	89.36
	19.90	162.90
	25.08	429.40

TABLE II.

Speeds.	(a)	(b)	(c)	(d)
12, 16, 20, 24 and 9, 12, 15, 18.	2.45946	1.4918	1.9962	2.0038
8, 12, 16, 20	3.21053	1.6487	1.9918	2.0082

TABLE III.

No.	Cube.	Fourth.
8	512	4,096
9	729	6,561
10	1,000	10,000
11	1,331	14,641
12	1,728	20,736
13	2,197	28,561
14	2,744	38,416
15	3,375	50,625
16	4,096	65,536
17	4,913	83,521
18	5,832	104,976
19	6,859	130,321
20	8,000	160,000
21	9,261	194,481
22	10,648	234,256
23	12,167	279,841
24	13,824	331,776
25	15,625	390,625

DISCUSSION.

REAR ADMIRAL JOHN LOWE, U. S. N., Retd.—Lieutenant Cox's paper is a very powerful and comprehensive article, but will provoke considerable dissent, for the reason that his equation is based upon the assumption of a *limiting speed*, the existence of which has not yet been proven or accepted.

In the prefatory portion of the paper acknowledgment is made of an article on the same subject, written by myself, and I am tacitly credited with the expression $y = k + cR^n$, for the curve of power and fuel consumption, in which

y = consumption of fuel, in tons per day of 24 hours;

k = a constant quantity of fuel consumed per day;

cR^n = fuel consumed in propelling a ship at the rate of R revolutions of the screw per minute.

The article then states that the values of n so deduced lie between 2 and 3; a remark to which I take exception, as follows:

After reading the text producing cR^n, it does not require extraordinary knowledge of physics to recognize in that term an expression for a power curve in which R is the variable. Now, in the nature of things, even when motion in the abstract is produced without friction and without turbulence, in an expression like cR^n for power, the value of n cannot be less than 3, and may be more, according to the amount of friction and turbulence involved in the transgression to imperfect practice from ideal theory.

Furthermore, even without this knowledge, when we consider the expression $y = k + cR^n$, we may contemplate a vessel in commission, but lying at her anchor. In this condition there is no propulsion, and, consequently, $y = k + o$; or, in ordinary language, the consumption is that due to banked-fire watches, maintained for distilling fresh water, for lighting the ship, for heating and ventilating, and other similar items. The amount of fuel thus consumed is definitely known by measurement, and is definitely recorded each day in the log.

It is plain, therefore, that the value of k is accurately known. Such being the case, let the vessel proceed under way for a definite time, at the rate of R revolutions of the screw per minute; the fuel then consumed is measured in the firerooms, and therefore y is accurately known. It is, therefore, manifest that the value of cR^n is definitely known, because $y - k = cR^n$.

It follows, therefore, that from two trials of the vessel, one trial at the rate of R revolutions, the other at the rate of R' revolutions, that the values of c and of n become manifest, because

$$\frac{\log (y - k) - \log (y' - k)}{\log R - \log R'} = n.$$

It may also be proven that n is not strictly a constant, but that it increases in value at the higher ordinates of the curve. Nevertheless, its value may be so chosen that the calculated curve produced by its use will accord very closely with the experimental curve produced by plotting the results of experimental trials. Any formula which will do this must be pronounced a good one.

It will be inferred from all this that I prefer the expression proposed by myself to the one produced by Lieutenant Cox, one reason being that it is simpler and more easily applied.

LIEUTENANT COMMANDER E. C. KALBFUS, U. S. N.—The only objection to Mr. Cox's method that I can find lies in the determination of the constants for completing the equation. Getting these constants under service conditions has proved to be difficult, although it is possible for a painstaking chief engineer to work up accurate data for his own ship. It seems to be impossible, except in a few instances, to supply such data from the records of the Department.

For the purpose of reducing the expenditure of any ship of a class to the average speed made by the class, as required by the present competition rules, this formula would be exactly what we need, provided the constants could be accurately determined. At present we are compelled to obtain the coal used for propulsion by subtracting from the total expenditure at any speed, the port consumption. This method is palpably inexact, but no more so than the use of a formula which contains incorrect or unreliable constants.

It may be possible, and I hope it will come to pass next year, that we will have sufficiently accurate data to draw the coal curve for the competing vessels, in which case this formula will be just what we want.

SOME THEORIES AND NOTES ON THE CORROSION OF COPPER AND BRASS.

By Lieutenant Commander W. B. Tardy, U. S. N., Member.

OBJECT OF PAPER.

The following theories and notes are chiefly a brief résumé of a paper read before the Institute of Metals by Mr. E. L. Rhead, F. I. C., and the discussion thereon. Nothing of value has been added, the object being to condense so that the various theories may be published in a service journal in such shape that the attention of all officers may be called to them, with the hope that those who have non-ferrous corrosion troubles and the opportunity of making tests may try out one or more with a view to our accumulated practical knowledge finally enabling us to arrive at a solution of the problem both as to cause and remedy.

In the above mentioned papers the discussion and recorded experiments tend to establish the following facts; that corrosion of copper and brass is due:

(1) "To methods of manufacture."

(2) "To selective chemical action."

(3) "Corrosion resulting from chemical action set up or promoted by the electrical conditions resulting from the presence, side by side, in the alloy, of different components containing the constituents of the alloy in varying proportions, or from the presence of included impurities."

(4) "The effect of vagrant electric currents escaping from the electrical equipment."

Without going into the details of the various experiments made, or into the highly technical reasoning, we will discuss in a popular way the above theories, in the order given.

It is conclusively shown that with copper of a fixed purity,

that part which is hardest, either from processes of manufacture or from operations in working it into shape, is more readily attacked and suffers more rapid deterioration. The evident remedy is uniform annealing, using copper or brass of the softest quality compatible with the service required of it. In this connection it was admitted in the discussion that copper corrosion troubles in serious form date from 1899, it being stated that this corresponds to the time when the first electrically-produced copper, manufactured in 1898, was put on the market. It is difficult to see how this method of producing copper would make it more susceptible to attack, provided, of course, that it is remelted afterwards. It would not differ from ordinary smelted copper, except that it might be somewhat purer. This immediately raises the question whether electrically-deposited copper has not had certain beneficial minute impurities removed. This question might, it seems to us, be very readily answered in a general practical manner by having two similar pipes installed close together in a ship to do the same duty; making all conditions identical except having one of the pipes made of electrolytic copper and the other copper purified to the same degree by smelting processes.

In regard to causes (2) and (3) it is stated that it does not seem possible for cause (2) to operate unless affected by (3), but that the electromotive force resulting from the contact of copper with zinc alloys containing copper in large amounts is very feeble, never exceeding .08 volt when immersed in sea water. A long series of experiments was made, immersing hard and soft copper and various copper alloys in different saline solutions, from which it is shown that the harder the metal the more susceptible it is to corrosion or to solution, that the relative order of the solution tensions of metals varies with the liquid in which the metal is placed; but that in every case the corrosive action is increased by the presence of carbonic-acid gas. In this connection the discussion brought forth descriptions of the rapid failure of condenser tubes due to lodgment of particles of cinder, ash and coal, or other forms of carbon in the tubes. The obvious manner of avoid-

ing troubles due to lodgment of gas-producing solids is to so locate suctions that they are well clear of ash discharges, have pipes, condenser tubes, etc., as straight and with as few pockets as possible, then increase the velocity of flow to the point where all silt or solid matter will be scoured out and swept through pipes or tubes.

The small electromotive-force difference above referred to (.08 volt) is the only force tending to produce electrolysis between elements of different composition, however slight, within the same pipe or tube, due to differences of metal, method of manufacture or working, or to slight local differences of alloy composition, if we except the effect of vagrant electric currents due to grounds on the ship.

If we accept the electrolytic theory of corrosion we must, to account for the rapid deterioration of copper pipes and copper alloys aboard ship, show a more powerful force setting up electrolytic decomposition and deposition than the insignificant potential difference (.08 volt, maximum) existing between a small copper element and an alloy element in the walls of a copper pipe. In general discussions, such as occur in ward-rooms, it is sometimes held that if grounds in the ship caused electrolytic action the steel, being electro-positive, would be attacked and the copper pipes would be immune or possibly have iron plated out on them. This is valid reasoning if one is justified in the assumption that each metal, the steel plating on the one hand and the copper pipe on the other, is so pure that all other possible electrolytic couples are thereby eliminated. A glance at Dr. Cushman's specimen of commercially homogeneous iron or steel imbedded in a culture rich in phenol-thalien as an indicator, which shows by color reaction what is taking place due to electro-chemical differences in the purest obtainable steel, shows that the above assumption is untenable.

It is shown in the discussion by the members of the Institute of Metals that the tubes of a condenser were protected and their life materially lengthened by making electric contact from the main injection pipe to the overboard discharge

pipe with a large capacity cable, thus electrically short circuiting the condenser. This was the main condenser of an electric-power plant in which there were known to be serious grounds.

From the above case, from our own experience and observation, and the experience of a large number of officers in the service, we offer the opinion that theory No. 4, vagrant electric currents escaping from the electric plant of the ship, furnish both the cause and the E.M.F. that produces by far the greater part of the rapid destructive copper, copper-alloy corrosion that occurs on board our ships. It is admitted at the outset that the fact that the ship is purposely made the ground for the wireless has no direct bearing or effect; since its E.M.F., and consequently its current is alternating, there can be no injurious effect other than the possibility of this alternating effect assisting a D.C. current to flow where otherwise the resistance was too great for it to be established or maintained by the small direct E.M.F. due to leaks from the plant.

It is a matter of common experience and observation that copper pipes most frequently fail at or near bends; one side or other, sometimes both sides, of flanges. It occurs so frequently next to flange at outboard end of line where suction pipe connects to stool pipe as to give rise among certain officers to the belief that the corrosion is, or may be, due to thermo-electric effects.

The location of the most destructive corrosion is where one would expect it, reasoning from the first three theories. The greatest differences in character of metal, due to working, are formed at bends and flanges. The greatest differences in composition are found near flanges due to brazing material, and the most active liberation of corrosive gases would naturally be at or near bends. Therefore postulate No. 4 is needed only to account for the rapidity of corrosion. The theory is that vagrant currents will take naturally the easiest path, that the copper pipe, where it is straight and continuous, offers this path, that at a flange the resistance is enormously in-

creased, due to discontinuity of pipe and interposition of gasket, so that the current finds its easiest path through the salt water by the joint. It leaves the pipe just short of one flange and reenters metal of pipe just beyond the joint; the point of leaving becoming the positive pole, the metal there going into solution and plating out further along the pipe or being swept out altogether. It is believed that some such action takes place at bends. So far as we know, nothing practical has ever been done to test this theory. The theory is given and the hope expressed that some engineer officer on a ship in commission will do something that will tend to prove or disprove it. With two similar or exact pipes doing as nearly as possible the same duty, one might have an uninsulated copper conductor of large capacity laid parallel, with one end in close mechanical contact with the pipe outboard of outboard flange and other end connected to pump or whatever pipe is attached to; or copper sleeve might be slipped over each joint, making electric contact with the pipe each side of flange, while no such precautions were taken with the similar pipe, long observation ought to give an indirect proof or determination.

With a Siemens-Hawlske thermocouple galvanometer and proper leads and contacts it ought to be possible to determine any potential differences from point to point in the body of a pipe, also the direction of current—at least such difference as would be produced by grounds—and from these determinations see whether corrosion does always, or usually, occur on the positive side of an interposed resistance.

SUMMARY.

Copper and brass corrosions are due probably to:

(1.) Methods of manufacture.

(2.) Selective chemical action.

(3.) Corrosion resulting from chemical action set up or promoted by electrical conditions resulting from the presence, side by side, in the alloy, of different components, containing

the constituents of the alloy in varying proportions, or from the presence of included impurities.

(4.) The effect of vagrant electric currents escaping from the electrical equipment.

TO PREVENT OR LIMIT CORROSION.

(1.) Uniformity, physical and chemical, in copper and copper alloys is necessary.

(2.) Prevent entrance of sewage, carbonic-acid gas, ashes or coal; if latter cannot be prevented, make pipes and tubes as straight and free of pockets as possible, and increase velocity of water so that they will be scoured out.

(3.) Eliminate or minimize grounds if the theory of vagrant currents be accepted.

If this be not accepted make tests on board ship to prove or disprove its validity; those suggested above, as well as others that will suggest themselves to the practical investigator.

DESCRIPTION AND TRIALS OF U. S. TORPEDO-BOAT DESTROYERS *SMITH* AND *LAMSON*.

By W. B. Robins.

Torpedo-boat destroyers numbers 17 and 18, the *Joseph B. Smith* and *Roswell H. Lamson*, were provided for by Acts of Congress approved June 29, 1906, and March 2, 1907, and were among the first five destroyers of the same date in the United States Navy to be equipped with turbine propelling machinery.

The contract for these vessels, signed October 10, 1907, called for delivery within 24 months, the contractors being The William Cramp & Sons Ship and Engine Building Company, Philadelphia, Penna. The contract time for the *Smith* was extended to November 24, 1909, and for the *Lamson* to January 27, 1910, the date in each case of delivery at the Navy Yard, Philadelphia. The *Smith* was launched April 20, 1909, and the *Lamson* June 16, 1909.

The general requirements of the contract were as follows:

Displacement, 700 tons.

Speed on four-hour full-speed trial, 28 knots.

Weight of machinery (including contained water), 275 tons.

Trials.—(a) A progressive trial over a measured-mile course in water not less than 30 fathoms deep for standardizing the screws, extending from maximum speed down to a speed of 12 knots; about twenty runs to be made over the course in order to adequately cover the range of speed desired.

(b) A full-speed (28 knots) four-hour trial, air pressure not to exceed an average of 5 inches of water, steam pressure at high-pressure turbine not to exceed 240 pounds above atmosphere, and water consumption not to exceed 15.5 pounds per H.P. per hour.

(c) A twelve-hour trial at 24 knots, water consumption not to exceed 17.3 pounds per horsepower per hour.

(d) A twenty-four hour trial at 16 knots, water consumption not to exceed 27.3 pounds per horsepower per hour. The requirement for the duration of this trial was reduced to 12 hours.

The horsepower used for computing water consumption on trials to be the shaft horsepower of the main turbines plus the indicated horsepower of all engine-room auxiliaries in use except the evaporating and distilling plant, the water for which is to be excluded from the measurement.

The contract price for each vessel was $585,000.00, of which $325,250.00 was allotted for machinery.

GENERAL DESCRIPTION OF THE VESSEL.

Hull.—The hull is of steel, the shell and frame galvanized below the water line. The sheer strake is of 15 pounds plate amidships, tapering to 8 pounds at the ends; the garboard strake is 13 pounds, tapering to 7 pounds at the ends; and the remainder of the shell plating is 10 pounds amidships, tapering to 7 pounds at the ends, except one strake, which is 9½ pounds tapering to 6 pounds. The main deck stringer plate is 17½ pounds amidships, tapering to 8 pounds at the ends, and the remainder of the main deck plating is 10 pounds, tapering to 6 pounds.

The main deck runs the full length of the vessel, a raised forecastle deck running about one-quarter of the length aft from the bow. A berth deck extends forward and aft of the machinery space.

On the forecastle deck are located the anchors and anchor davit, capstan, a 3-inch semi-automatic gun on the center line, pilot house with bridge on top, and two .30-caliber automatic guns. The steering engine is located in the pilot house.

The space on the main deck, under the forecastle, contains the officers' quarters, galley, crew's wash room, lamp room, and paints and oils room; in the latter is located the windlass engine.

On the main deck, immediately abaft the forecastle, two 3-inch semi-automatic guns are placed, one on each side. Abaft these are stowed four small boats.

Amidships on the main deck are two torpedo tubes, one on each side, with boxes for stowing two torpedoes just forward of the engine hatch. Abaft the engine hatch there is a 3-inch semi-automatic gun on the center line of the vessel, and a small deck house containing crew's toilet and wash room and a companionway to the berth deck. Abaft the deck house is placed a torpedo tube on the center line of the vessel, the torpedo for this tube being stowed in a box alongside the deck house.

On the main deck away aft are a 3-inch semi-automatic gun on the center line, the tiller, and a hand steering gear.

On the berth deck forward and aft are storerooms and quarters for crew, chief petty officers and petty officers. Below the berth deck, forward of the machinery space, are trimming tanks, chain locker, storerooms, fresh-water tanks, magazines, handling room, and reserve feed-water tank. Below the berth deck aft are reserve feed-water tank, magazine for war heads, handling room, magazine, shaft alleys, storerooms, and trimming tanks.

The machinery and coal space occupies nearly one-half the length of the vessel. There are two boiler rooms with coal bunkers outboard, and one engine room the full width of the vessel. There is also a coal bunker between the forward and after boiler rooms, and one between the after boiler room and the engine room.

Particulars of Hull.

Length between perpendiculars, feet and inches	289–00
over all, feet and inches	293–10½
on load water line, feet and inches	289–00
of straight keel, feet and inches	243–05½
Breadth, molded, extreme, feet and inches	26–04½
extreme, over plating, feet and inches	26–05½
over guards, feet and inches	27–00
Depth, molded, main deck at side, M. S. No. 82½, feet and inches,	16–04½

Draught above bottom of keel, feet and inches	8–00½
Ratio of length to beam at L.W.L.	11.1026
Displacement, normal, 8 feet 0½ inch draught, tons	700.0
per inch at 8 feet 0½ inch draught, tons	11.85
Area of immersed midship section, square feet	135.3
L.W.L. plane, square feet	4,982.0
wetted surface, square feet	7,382.0
C. G. of L.W.L. plane, aft of M. S., No. 82½, feet	1.8
C. B. above bottom of keel, feet	5.14
forward of M. S., feet	0.13
Transverse metacenter above C. B., feet	8.12
Longitudinal metacenter above C. B., feet	822.0
Coefficient of fineness, block	0.407
midship section	0.649
L.W.L.	0.662
cylindrical	0.627
Number of frames, spaced 21 inches throughout	166
watertight compartments	28

Heights above 8 feet 0½ inch W.L.

Foremast, feet and inches	60–09⅛
Mainmast, feet and inches	60–06⅞
Bridge at center, frame 37, top of beam, feet and inches	24–04⅝
side, frame 37, top of beam, feet and inches	23–10⅝
Top of forecastle deck at stem, feet and inches	17–03⅞
Main deck at stern, feet and inches	8–03⅝
Height of bridge at center above forecastle deck, feet and inches,	7–06

Capacities of Fresh-Water Tanks.

Compartment.	Gallons.	Tons F. W.
A–6	3,494.06	12.98
Gravity tank	50.0	0.19

Capacities of Reserve-Feed Tanks.

Compartment.	Gallons.	Tons F. W.	Tons S. W.
A–10	1,966.72	7.3	7.51
D–1	1,160.9	4.31	4.42

Capacities of Trimming Tanks.

Compartment.	Gallons.	Tons F. W.	Tons S. W.
A–1	508.05	1.88	1.93
A–2	2,734.69	10.16	10.45
D–6	2,170.25	8.06	8.29
D–7	1,291.14	4.79	4.93

Small Boats.

2 20-foot whale boats.
1 17-foot dory.
1 20-foot 10-inch power dory.

Battery.

3 5-meter × 45-centimeter deck torpedo tubes.
5 3-inch (14-pounder) semi-automatic 50-caliber guns.
2 .30-caliber automatic guns.

COMPLEMENT.

Commanding officer	1
Wardroom officers	3
Seamen's branch	27
Engine-room branch	51
Special branch	2
Commissary branch	2
Messmen's branch	3
Total	89

MACHINERY.

Main Turbines.—There are five ahead turbines and two backing turbines, all of the Parsons type, arranged on three shafts. The main high-pressure turbine is on the center shaft; the intermediate cruising, one low-pressure and one backing turbine on the starboard shaft; the high-pressure cruising, one low-pressure and one backing turbine on the port shaft; the backing turbines being enclosed in the low-pressure casings. The turbines were designed to run at about 775 r.p.m. at full speed with steam at 250 pounds per square inch at the main high-pressure turbine, the equivalent I.H.P. to be about 10,000.

The turbine arrangement, blading of rotors and combinations for different speeds are as shown on plate I.

The turbine casings are all of cast iron, divided in halves along their horizontal axes, the upper halves being removable for examination and repair of blading, etc.

= CONDITIONS =

No I.	Maximum full speed ahead.	Live steam at 250 lbs. to Main H.P. and bypass, then at about 35 lbs. to both L.P's.
No II.	Reduced full speed ahead.	Live steam at 250 lbs. to Main H.P. with by-pass closed, then at about 35 lbs. to both L.P's.
No III.	High cruising speed ahead.	Live steam at 250 lbs. to I.P. cruising, then at 90 lbs. to Main H.P. then at about 40 lbs. to both L.P's.
No IV.	Low cruising speed ahead.	Live steam at 250 lbs. to H.P. cruising, then at 90 lbs. to I.P. cruising, then at 30 lbs. to Main H.P. then at 8 lbs. to both L.P's.
No V.	Manoeuvring speed ahead.	Live steam at 250 lbs. to both L.P's only.
No VI.	Backing.	Live steam at 250 lbs. to Astern turbines in after end of L.P. casing.

SKETCH SHOWING ARRANGEMENT OF TURBINES AND BLADING OF ROTORS AND STEAMING CONDITIONS TORPEDO BOAT DESTROYERS Nos. 17 & 18

Note: Number of blades is approximate.

OFFICE OF INSPECTOR OF MACHINERY,
FOR U. S. NAVY
The William Cramp & Sons Ship & Engine
Building Works, Philadelphia
P.P.B., September 23 1908

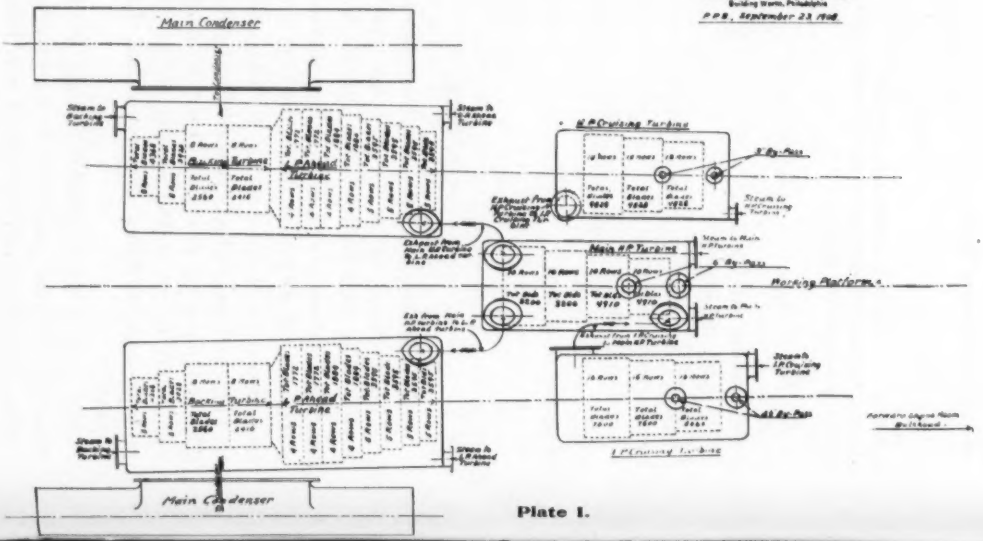


Plate I.

The rotors consist of forged-steel drums, cast-steel end wheels and forged-steel spindles. The end wheels are shrunk on and pinned to the spindles and the drums shrunk on and pinned to the end wheels. The blading is all of rolled brass of special composition, being held in place in grooves in the cylinders and rotors by packing pieces which are caulked into small grooves in the sides of the main grooves. The blades, except some of the shortest lengths, are stiffened by a binding strip passing around each row near the ends of the blades, the strip being laced to the blades by copper wire soldered on with silver solder.

The dummy rotors are steel castings secured to the rotor drums and running in cast-iron dummy rings bolted to the turbine casings. The dummy packing is of the usual Parsons labyrinth type. The designed dummy clearances in a fore-and-aft direction are as follows: Main H.P., .015 inch; L.P., .02 inch; H.P. cruising, .015 inch; I.P. cruising, .015 inch. The maintenance of the proper dummy clearance being an important feature in the operation of this type of turbine, a micrometer is fitted to each turbine for measuring this clearance. The micrometers work through stuffing boxes in the turbine casings, so that the clearances may be determined while the turbines are running.

Turbine Cylinder Data.

Cylinder.	Rotor drum.		Diameter of cylinder for each stage.								Number of rows of blades for each stage.							
	Dia.	Lgth.	1st.	2d.	3d.	4th.	5th.	6th.	7th.	8th.	1st.	2d.	3d.	4th.	5th.	6th.	7th.	8th.
	ins.	*ins.*	*ins.*	*ins.*	*ins.*	*ins.*	*ins.*	*ins.*	*ins.*	*ins.*								
H.P. cruising...	30	50⅜	30⅜	31	31⅜	...	...	...	...	...	18	18	18	...	...	...	...	...
I.P. cruising.....	29	58⅛	30⅜	31¼	32	...	...	...	...	...	16	16	16	...	...	...	...	...
Main H.P.....	30	62¼	32⅜	34	35⅛	38	...	...	...	...	10	10	10	10	...	...	...	...
Main L.P., stb. and port.....	44	62	46⅜	48	49½	52	54	59	59	59	5	5	5	5	4	4	4	4
Astern, starbd. and port	30	57½	31½	32½	35	35	...	...	...	...	8	8	8	8	...	...	...	...

Turbine Drains.—From the bottom, near each end of the main H.P. and cruising turbine casings, 1-inch drain pipes lead to manifolds from which 2-inch pipes lead to drain pock-

ets at bottom of the after ends of the L.P. turbines. From these pockets the water is drawn, through 2½-inch pipes with check valves, by the main air pumps.

Turbine Relief Valves.—On the exhaust end of the main H.P. turbine and each cruising turbine there is a 2½-inch spring-loaded relief valve, the valve on the main H.P. being set at 50 pounds and those on the cruising turbines at 110 pounds. On the inlet end of each L.P. turbine there is a 3½-inch spring-loaded relief valve set at 50 pounds.

Shafting.—Each system of shafting comprises the rotor spindles, two sections of line shaft and one propeller shaft. All shafting, except the rotor spindles, is solid. The thrust collars are on the forward rotor spindles. The couplings between the low-pressure and cruising-turbine rotor shafts are of special design, allowing independent fore-and-aft movement of the shafts to provide for expansion and to permit the adjustment of the rotors for dummy clearance. The couplings at the forward ends of the propeller shafts are of the removable disc and split-collar type to allow the shafts to be withdrawn outboard. All other shaft couplings are of the usual flange-and-bolt type.

Particulars of Shafting.

Rotor shafts:	Forward.	After.
Main H.P. turbine.		
Diameter, inches	9 and 7	9 and 7
of hole, inches	5¼ and 3	5¼ and 3¼
Length, feet and inches	4-6¾	3-4$\frac{11}{16}$
L.P. and astern turbines.		
Diameter, inches	9 and 7	9 and 7
of hole, inches	5¼ and 3	5¼ and 3¼
Length, feet and inches	4-9¾	3-6$\frac{11}{16}$
H.P. cruising turbine.		
Diameter, inches	8¼ and 6¼	8¼ and 6¼
of hole, inches	5 and 3¼	5 and 3¼
Length, feet and inches	3-8$\frac{13}{16}$	3-0⅝
I.P. cruising turbine.		
Diameter, inches	8¼ and 6¼	8¼ and 6¼
of hole, inches	5 and 3¼	5 and 3¼
Length, feet and inches	3-8$\frac{15}{16}$	3-2⅝

	Center.	Wing (each).
Line shafts:		
Number	2	2
Diameter, inches	$6\frac{1}{4}$	$6\frac{1}{4}$
Length, feet and inches	$23\text{-}5\frac{7}{16}$	$\begin{cases} 10\text{-}0 \\ 14\text{-}6\frac{5}{8} \end{cases}$
Diameter of coupling flanges, inches	$11\frac{1}{2}$	$11\frac{1}{2}$
Thickness of coupling flanges, inches	$1\frac{1}{2}$	$1\frac{1}{2}$
Number of taper coupling bolts, $1\frac{5}{8}$ inches mean diameter	8	8
Propeller shafts:		
Number	1	1
Diameter, inches	$6\frac{1}{2}$	$6\frac{1}{2}$
Length, feet and inches	$42\text{-}3\frac{1}{4}$	$42\text{-}3\frac{1}{4}$
Diameter of threaded end for propeller nut, inches	5	5
Thickness of brass sleeves at bearings, ins.	$0\frac{7}{16}$	$0\frac{7}{16}$
Length of shafting from flange at after end of rotor shaft, feet and inches	$89\text{-}1\frac{1}{4}$	$66\text{-}9$

	Starboard.	Center.	Port.
Total length of shafting, including rotors, feet and inches	$94\text{-}11\frac{7}{8}$	$101\text{-}8\frac{1}{4}$	$94\text{-}1\frac{7}{8}$

Bearings.—Each turbine has two main bearings, one at each end, with a thrust bearing at the extreme forward end. Each main bearing consists of two composition babbitt-lined half-brasses and a cast-iron cap, the bottom brass being set in an extension of the cylinder casting. Each thrust bearing consists of two forged-steel half thrust blocks, and a number of composition half rings as given below, and a cast-iron cap, the lower half thrust block being set in an extension of the cylinder casting. The lower half thrust block takes the ahead thrust and the upper the astern thrust, when there is any thrust; the dummies being so proportioned that when the turbines are running at their designed speed and power the steam thrust is approximately equal to the propeller thrust.

The thickness of the thrust rings is 1/16 of an inch less than the space between the collars on the shaft, and the two half thrust blocks are adjusted in fore-and-aft relation to each other so that the rings in the lower half bear against the forward sides, and those in the upper half against the after sides of the thrust collars on the shaft.

The center line shafting has three steady bearings and two

stern-tube bearings. Each line of wing shafting has one steady bearing, two stern-tube bearings and one strut bearing. The steady bearings and their caps are cast iron, babbitt lined, and all are fitted with stuffing boxes to prevent the escape of oil. The stern-tube and strut bearings are of the usual naval type, of composition lined with lignum-vitae.

Particulars of Bearings.

	Main H.P.	H.P. cruising.	I P. cruising.	L.P. and astern.
Rotor shaft bearings:				
Number, each	2	2	2	2
Diameter, inches	7	6½	6½	7
Length, inches	7	6½	6½	12
Thrust bearings:				
Collars on shaft.				
Number	13	7	7	13
Thickness, inch	0 9/16	0½	0½	0 9/16
Space between, inch	0 1/16	0⅝	0⅝	0 1/16
Outside diameter, inches	9½	8⅝	8⅝	9½
Inside diameter, inches	5¾	5¾	5¾	5¾
Bearing surface, square inches	506.0	211.0	211.0	506.0
Thrust rings:				
Number, ahead	13	7	7	13
astern	12	6	6	12
Thickness, inch	0⅝	0 9/16	0 9/16	0⅝
Outside diameter, inches	10¾	9⅞	9⅞	10¾
Inside diameter, inches	6	6	6	6
Effective surface, ahead, sq. in	253.0	105.5	105.5	253.0
astern, sq. in	233.0	90.0	90.0	233.0
Steady bearings:			Wing.	Center.
Number			1	2
Diameter, inches			6½	6½
Length, inches			11¾	11¾
Number			...	1
Diameter, inches			...	6½
Length, inches			...	11¾
Stern-tube bearings:				
Forward bearing.				
Diameter, inches			7⅞	7⅞
Length, inches			28 1/16	28 1/16
After bearing.				
Diameter, inches			7⅞	7⅞
Length, inches			28 1/16	36 3/16
Strut bearing.				
Diameter, inches			7⅞	...
Length, inches			36 3/16	...

PROPELLERS.

The three propellers are of the three-bladed, true-screw type, and are solid manganese-bronze castings, the faces of the blades being machined. The propellers were measured after being machined and were found to be true screws.

Propeller Data.

	As used on official trial.
Number of blades, each	3
Diameter, feet and inches	4–10
Pitch, feet and inches	5–00
Projected area, square feet	11.38
Helicoidal area, square feet	12.65
Disc area, square feet	18.35
Ratio, pitch to diameter	1.034 : 1
projected area to disc area	0.620 : 1
Thickness of blade at root, inches	0–02⅜
Width of key (two for each propeller) inches	0–01½
Depth of key, inch	0–00¾
Height of lower tip of blade above keel, wing, inches	0–13
Depth of lower tip of blade below keel, center, inches	0–11
Immersion of upper tip of blade at 8 feet 0½ inch draught, wing, ins.	0–25½
center, ins.	0–49½
Center of center propeller aft of wing propellers, feet and inches	8–03
below wing propellers, feet and inches	2–00

On the builders' first trial of the *Smith* it was found that the revolutions of the turbines were below the designed rate, and as the slip was also moderate, the area of the propellers was reduced by taking one inch off the tip of each blade, making the diameter 4 feet 10 inches instead of 5 feet. This alteration was also made to the propellers of the *Lamson*.

MAIN CONDENSERS.

There are two main condensers of oval, flat-sided section with the long axis vertical, located outboard alongside the low-pressure turbines. The shells are of steel plate, stiffened by angles and stay rods. The heads are of sheet copper, with brass flanges. The tube sheets are of rolled naval brass and the tubes of composition: copper, 70; tin, 1; zinc, 29. The tubes are packed at the tube sheets with special cotton-tape

packing held in place by screwed ferrules, and are supported at two intermediate points by steel diaphragms with brass ferrules through which they pass.

Particulars of One Condenser.

Length between tube sheets, feet and inches	14–00
Height of shell, inside, feet and inches	6–07⅜
Breadth of shell, inside (mean), feet and inches	2–05¾
Thickness of shell plates, sides, inch	00 7/16
top and bottom, inch	00½
tube sheets, inch	01
Number of tubes	1,752
Outside diameter of tubes, inch	00⅝
Thickness of tubes, B.W.G., number	18
Cooling surface, square feet	4,013
Diameter of circulating-water inlet, inches	15
outlet, inches	15
Length of rectangular exhaust-steam inlet, feet and inches	6–05
Breadth of rectangular exhaust-steam inlet, feet and inches	2–00
Diameter of main air-pump suction nozzle, inches	09½

MAIN AIR PUMPS.

There are two main air pumps of the Blake vertical-beam type, each pump having one steam cylinder and two pump cylinders. They are located at the after end of the engine room, close to the center line of the ship. The pump cylinders are of the bucket type.

Particulars of One Main Air Pump.

Diameter of steam cylinder, inches	10
pump cylinders, inches	22
Stroke of all pistons, inches	15
Diameter of suction nozzle, inches	09½
discharge nozzle, inches	06½
steam pipe, inches	01½
exhaust pipe, inches	01½

MAIN CIRCULATING PUMPS.

There are two single-inlet centrifugal circulating pumps driven by single-cylinder vertical steam engines. They are located at the outboard sides of the engine room, forward of the condensers.

Particulars of Main Circulating Pump.

Diameter of impeller, inches	26
Width of impeller at tip of blades, inches	03½
Diameter of suction nozzle, inches	15½
discharge nozzle, inches	15
steam cylinder, inches	07
Stroke of steam piston, inches	06
Diameter of steam pipe, inches	01½
exhaust pipe, inches	02

BOILERS.

There are four Mosher water-tube coal-burning boilers arranged in two firerooms, each boiler having its own smoke pipe. Each boiler has one cylindrical steam drum and two semi-cylindrical water drums. The steam drum is made of two plates, the plate forming the lower half, into which the tubes are expanded, being 1⅛ inches thick, and the plate forming the upper half being 9/16 of an inch thick. The inner plates or tube sheets of the water drums are 1 inch thick and the outer plates are 7/16 of an inch thick.

The tubes are all slightly curved, so that all may be withdrawn through a number of holes in the upper part of the steam drum, these holes being closed by steel bolts having countersunk heads inside the drum bearing on copper ferrules, and nuts with steel washers on the outside of the drum. The tubes of the inner row on each side are bent between those of the next row, forming a wall on each side of the fire extending from the back of the grate to within 2 feet 3 inches of the front.

The flow of the products of combustion is through the openings at the front of the tube walls, then back among the tubes to the uptake at the back of the boiler. The feed water enters the steam drum at the front end near the bottom and passes through two 2-inch brass pipes to the rear part of the drum where it is discharged through perforations in the pipes. The normal water level is at the center of the steam drum.

The boiler casings consist of an outer casing of No. 16 U. S. S. G. steel plate and an inner casing of No. 14 U. S.

S. G. steel plate, between which is 1 inch of asbestos felt. The furnaces are lined at the front and back and at the sides up to the water drums with fire brick $2\frac{1}{2}$ inches thick. The air for combustion enters through doors on all four sides of the ash pan. There is an air space between the casing and the fire-brick lining at the front and back of the furnace, from which air is admitted through holes to the furnace above the fire, this air space being open to the ash pan at the bottom. The grate bars are of cast iron in three lengths for each boiler. There are four 16-inch by 18-inch furnace doors on each boiler.

Particulars for One Boiler.

Length of steam drum, feet and inches	10–10 $\frac{1}{16}$
Inside diameter of steam drum, inches	0–42
Thickness of steam drum, tube sheet, inches	0–01 $\frac{1}{8}$
top plate, inch	0–00 $\frac{9}{16}$
front head, inch	0–00 $\frac{5}{8}$
back head, inch	0–00 $\frac{9}{16}$
Radius of curvature of steam-drum heads, inches	0–42
water drum (semi-cylindrical), inches	0–15 $\frac{5}{8}$
Thickness of water-drum tube sheet, inch	0–01
outer plate, inch	0–00 $\frac{7}{16}$
front head, inch	0–00 $\frac{1}{2}$
back head, inch	0–00 $\frac{7}{16}$
Radius of curvature of water-drum heads, inches	0–31 $\frac{1}{4}$
Number of tubes (seamless, cold-drawn steel)	2,000
Outside diameter of tubes, inches	0–01 $\frac{1}{4}$
Thickness of tubes B.W.G., number	12
Length of longest tube, feet and inches	7–06 $\frac{7}{8}$
Radius of curvature of longest tube, feet and inches	14–08
Length of shortest tube, feet and inches	6–06 $\frac{3}{4}$
Radius of curvature of shortest tube, feet and inches	200–00
Heating surface, one boiler, square feet	4,500.85
Total heating surface, four boilers, square feet	18,003.40
Length of grate, feet and inches	8–04 $\frac{1}{2}$
Width of grate, feet and inches	11–00
Per cent. of air space in grate	49
Grate surface, one boiler, square feet	92.125
Total grate surface, four boilers, square feet	368.5
Ratio of heating surface to grate surface	48.86 : 1
Area of smoke pipe, square feet	13.45
Ratio of grate surface to smoke-pipe area	6.85 : 1

External length of boiler, feet and inches	10–10⅝
width of boiler, feet and inches	15–03½
height of boiler, feet and inches	11–07⅛
Designed working pressure, pounds per square inch	250.0
Weight of one boiler complete, dry, pounds	41,818.0
tons	18.67
with water at steaming level and temperature, pounds	52,042.0
with water at steaming level and temperature, tons	23.23

Each boiler is provided with the following fittings:

1 1¼-inch fire extinguisher.

2 2-inch internal feed pipes.

1 2½-inch combined main and auxiliary feed stop and check valve.

1 6½-inch dry pipe.

1 6½-inch steam stop valve.

1 1¼-inch internal surface-blow pipe with scum pan.

1 1¼-inch surface-blow valve.

2 1¼-inch internal bottom-blow pipes in water drums.

2 1¼-inch bottom-blow valves.

34 12-inch by 6-inch by ½ inch zincs.

1 4-inch duplex spring-loaded safety valve.

2 Klinger reflex water gauges, 10¼-inch, exposed length.

3 try cocks.

2 1-inch drain cocks.

1 steam gauge, 6-inch dial.

There is a system of pipes for blowing soot from the boiler tubes, these pipes having connections for the use of either steam or compressed air.

UPTAKES AND SMOKE PIPES.

The uptakes are made of two thicknesses of plate with 2½ inches of air space between. There are four smoke pipes of flat-sided oval section, the long axis being fore-and-aft. The smoke pipes are raked aft 4 inches, the rake beginning about 9 inches above the deck.

Particulars of Smoke Pipes.

Long diameter, inside, feet and inches	4–09
Short diameter, inside, feet and inches	3–04
Long diameter, outside, feet and inches	5–10
Short diameter, outside, feet and inches	3–10
Height above base line, feet and inches	29–06
grate, feet and inches	24–07½
Area of cross section, inside, square feet	13.45
Ratio of grate surface to smoke-pipe area	6.85 : 1

Owing to extreme discomfort at times during trials, caused by large quantities of soot and cinders from the fires falling on deck and in hatches, the height of the smoke pipes was increased eight feet after the vessel was delivered.

MAIN STEAM PIPING.

All main steam piping is of seamless-drawn steel with wrought-steel flanges. For sizes up to 5½ inches the pipes are rolled into the flanges, each flange having three grooves and a slight counterbore. For sizes 6-inch and above the pipes are welded to the flanges.

There is a 6½-inch branch leading from the stop valve on each boiler. The branches from the two forward boilers unite in one 9-inch pipe running aft along the starboard side to the engine-room bulkhead. The branches from the two after boilers unite in one 9-inch pipe running aft along the port side to the engine-room bulkhead.

All the valves controlling the steam to ahead and astern turbines are grouped at the working platform at the extreme forward end of the engine room according to the arrangement shown on Plate II. There are two 9-inch stop valves taking steam from the fireroom mains. From each of these valves, there is a branch leading to the 12-inch main throttle valve and a branch leading to the maneuvering and cruising-turbine valves. The steam from the 12-inch throttle valve passes through a strainer of perforated sheet brass and divides into two 8½-inch pipes leading to the main high-pressure turbine.

On the branch from the starboard 9-inch main stop valve there are the following valves: 8-inch steam to astern turbine; 6-inch maneuvering steam to L.P. ahead turbine; 8-inch

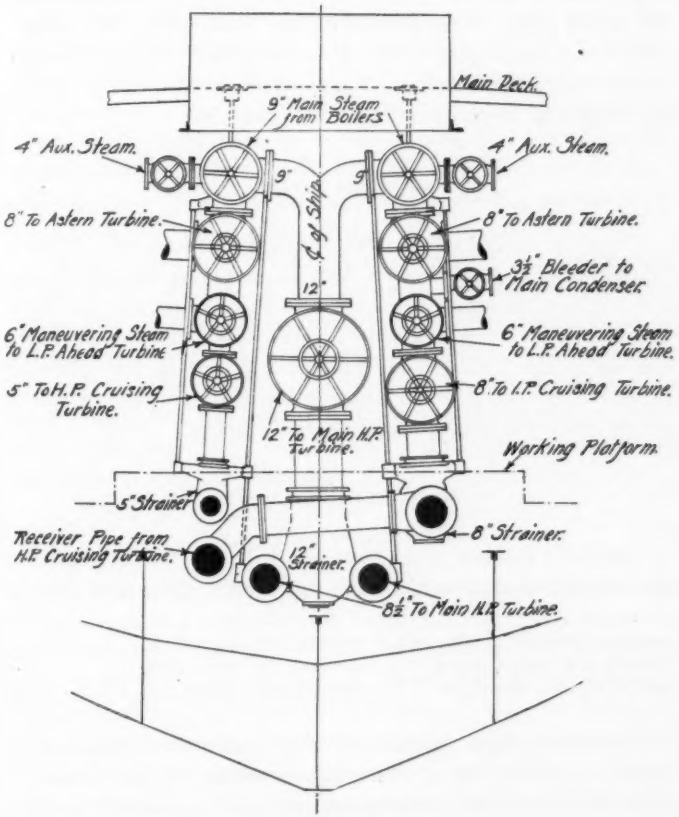


Plate II.

ARRANGEMENT OF VALVES AT WORKING PLATFORM.

steam to I.P. cruising turbine (high cruising speed); and 3½-inch bleeder to main condenser.

On the branch from the port 9-inch main stop valve there are the following valves: 8-inch steam to astern turbine; 6-

inch maneuvering steam to L.P. ahead turbine, and 5-inch steam to H.P. cruising turbine (low cruising speed).

All throttle and maneuvering valves are fitted with extra-large hand wheels to insure ease and rapidity of operation. All steam pipes to turbines are fitted with perforated sheet-brass strainers to prevent the entrance of any substance which might injure the blading. These strainers also act to a certain extent as separators to dry the steam.

RECEIVER PIPING.

From the main high-pressure turbine there are two 16-inch branches with spring-loaded non-return valves leading to the low-pressure turbines. From the H.P. cruising turbine there is a 9-inch receiver pipe with a screw-down non-return valve leading to the I.P. cruising turbine. From the I.P. cruising turbine there is a 13-inch receiver pipe with screw-down non-return valve leading to the main H.P. turbine.

MAIN EXHAUST PIPES.

The main exhaust pipes from the L.P. turbines to condensers are rectangular trunks, built up of steel plates and shapes.

Length of exhaust opening in L.P. turbine, feet and inches	3-11
Breadth of exhaust opening in L.P. turbine, feet and inches	3-03
Area of exhaust opening in L.P. turbine, square inches	1,833

As an indication of the extent to which the expansion of steam is carried out in this type of turbine, it may be of interest to compare the ratio of area of main steam inlet to main exhaust outlet with the corresponding ratio in a reciprocating engine for the same class of vessel. In the U. S. S. *Bainbridge*, one of the latest destroyers having reciprocating engines, the ratio of main steam inlet to main exhaust outlet is 1 : 8, while in the *Smith* and *Lamson* the corresponding ratio is 1 : 32.

AUXILIARY STEAM PIPING.

The auxiliary steam pipe in the engine room consists of a loop of 3½-inch pipe leading completely around the engine room, with branches to various auxiliaries, steam supply being taken from connections below the seats of the 9-inch main steam stop valves at the forward end of the engine room. In each fireroom there is a 2½-inch auxiliary steam pipe leading athwartship, with branches to the various auxiliaries, taking steam from the main steam branch from the forward boiler in each fireroom. There is also a 2-inch connecting pipe between the auxiliary steam systems of the two firerooms.

Steam Piping to Turbine Glands.—Steam for the turbine glands is supplied by a 1½-inch branch from the auxiliary steam line. There is a reducing valve at the connection from the auxiliary steam line reducing the pressure to two pounds per square inch. There is a separate branch with valve to each gland, and the main supply pipe is fitted with a relief valve at its after end, with branches discharging to the fifth expansion of the L.P. turbines.

AUXILIARY EXHAUST PIPING.

The exhaust-pipe branches from the various auxiliaries lead into an auxiliary exhaust main running through the firerooms and continuing in a loop around the engine room. In the engine room there are connections for directing the auxiliary exhaust into the starboard main condenser, the port L.P. turbine, the auxiliary condenser, or the feed-water heater. The dynamo turbine has an independent exhaust to the starboard main condenser. The auxiliary exhaust line has a connection to the escape pipe in the after fireroom for exhausting to the atmosphere.

WHISTLE AND ESCAPE PIPES.

There is a 5½-inch escape pipe on each smoke stack, leading from the safety valve on the boiler, the after escape pipe having a connection from the auxiliary exhaust. On the forward

side of the forward smoke stack there is a whistle with 6-inch diameter bell and 1½-inch steam connection.

MAIN INJECTION AND OUTBOARD DELIVERY PIPING.

Each main circulating pump draws from the sea through a 15-inch angle valve and discharges through a 15-inch copper pipe to the forward end of the main condenser. There is also a 7-inch valve on each main circulating pump with suction pipe from the engine-room bilge. At the after end of each main condenser there is a 15-inch copper pipe leading overboard through a 15-inch gate valve.

MAIN AIR PUMP SUCTION AND DISCHARGE.

Each main air pump draws from its condenser through a 9½-inch pipe with gate valve at the pump. There is a 6-inch cross-connection pipe with gate valve between the suction pipes from the two condensers. There is a 1-inch connection from the reserve feed-tank suction pipe to the port main air-pump suction for make-up feed. There is a 2½-inch connection from turbine cylinder drains on each main air-pump suction. Two 6½-inch discharges from the air pumps unite in one 9-inch pipe leading to the main feed tank. This pipe was fitted with temporary connection to the feed-measuring tank on deck for use during the trials.

FEED PIPING.

There is a feed-suction pipe from the main feed tank at the after end of the engine room, leading through the engine room and both firerooms. This pipe is 6½ inches in diameter from the feed tank as far as the auxiliary feed pump in the after fireroom, forward of which it is 4½ inches in diameter. There is a 3-inch connection at each end of the feed-suction line from the forward and after reserve feed tanks respectively. Each main feed pump has a 6-inch suction from the feed-suction line and the after main feed pump has also a 2½-inch suction from the main air-pump suction pipe.

Two 4½-inch discharges from the main feed pumps unite in one 5-inch pipe leading to the firerooms, with connections for discharging through the feed heater or by-passing direct to the boilers. The main feed line is 5 inches in diameter as far as the branch to the after boilers, forward of which it is 3½ inches in diameter. The branch to each boiler is 2½ inches in diameter. Each auxiliary feed pump has a 4½-inch suction from the feed-suction line and a 3½-inch discharge, dividing into two 2½-inch branches to the boilers in its compartment.

FEED-WATER HEATER.

At the forward end of the engine room, close to the main feed pumps, there is a cylindrical feed-water heater. The feed water passes through the tubes and the exhaust steam is admitted to the shell around the tubes. The shell is built up of steel plates, the heads of cast steel and the tubes and tube sheets of brass. The tubes are expanded into the tube sheets and expansion is provided for by a copper ring joining the two parts of the shell. The drain from the feed-heater shell has connections to port main condenser, auxiliary condenser and feed tank.

Particulars of Feed-Water Heater.

Diameter of shell, inside, inches	23
Thickness of shell, inches	00 5/8
Length between tube sheets, feet and inches	6-00
Diameter of tubes, outside, inches	00¾
Thickness of tubes, B.W.G., number	18
Number of tubes	508
Thickness of tube sheets, inches	01¼
Diameter of feed-water inlet, inches	05
outlet, inches	05
exhaust-steam inlet, inches	06½
drain outlet, inches	02½
Heating surface, square feet	499

FEED PUMPS.

There are two main feed pumps on the port side near the forward end of the engine room. They are of the Blake vertical simplex piston type.

Particulars of Main Feed Pump.

Diameter of steam cylinder, inches	14½
pump cylinder, inches	10
Stroke of pistons, inches	10
Diameter of suction nozzle, inches	6
discharge nozzle, inches	4½
steam pipe, inches	2
exhaust pipe, inches	2½

There are two auxiliary feed pumps, one in each fireroom. They are of the same dimensions as the main feed pumps, except that the suction and discharge nozzles are 4½ and 3½ inches in diameter, respectively. On the delivery side of each main and auxiliary feed pump there is a 2½-inch spring-loaded relief valve, discharging into the pump suction.

FEED AND FILTER TANK.

There is a galvanized sheet-steel feed tank of 650 gallons capacity in the after end of the engine room, the upper part being fitted as a filter with two filter compartments through which the water flows successively.

FEED MEASURING APPARATUS.

During the trials the discharge from the main air pumps was led to a calibrated measuring tank on deck, from which it was discharged by gravity to the feed tank in the engine room. The measuring tank was made with two compartments of 2,745 pounds capacity each, and the discharge pipe from the air pumps was fitted with a swivel connection allowing the flow to be directed alternately into the measuring compartments, each compartment being provided with a discharge valve at the bottom.

ENGINE ROOM FIRE AND BILGE PUMP.

In the forward part of the engine room, on the starboard side, there is a Blake vertical simplex piston fire and bilge pump. This pump has connections to draw from the sea, the

engine-room bilge and the drainage system, and to discharge to the sea, the fire main and the distiller circulating system. There are 1½-inch hose connections on both suction and discharge, and a 1½-inch spring-loaded relief valve on the discharge.

Particulars of Engine-Room Fire and Bilge Pump.

Diameter of steam cylinder, inches	10
pump cylinder, inches	7
Stroke of pistons, inches	8
Diameter of suction nozzle, inches	4
discharge nozzle, inches	3½
steam pipe, inches	1½
exhaust pipe, inches	1½

FIREROOM FIRE AND BILGE PUMP.

There are two Blake vertical duplex piston-type fire and bilge pumps, one in each fireroom. These pumps have connections to draw from the sea and the fireroom bilges, and to discharge to the fire main, the ash ejectors and overboard. Each pump has 1½-inch hose connections on both suction and discharge, and a 1½-inch spring-loaded relief valve on the discharge.

Particulars of Fire-Room Fire and Bilge Pump.

Diameter of steam cylinders, inches	7½
pump cylinders, inches	5
Stroke of pistons, inches	8
Diameter of suction nozzle, inches	4
discharge nozzle, inches	3½
steam pipe, inches	1½
exhaust pipe, inches	2

AUXILIARY CONDENSER AND PUMPS.

At the after end of the engine room, on the starboard side, there is a horizontal cylindrical auxiliary condenser, having 254 square feet of cooling surface. The air and circulating pumps are of the Blake horizontal combined type, located under the condenser. The circulating pump draws from the auxiliary sea valve and discharges through the condenser into

the main outboard delivery pipe. The air pump draws from the condenser and discharges into the discharge pipe from the main air pumps to the feed tank.

Particulars of Auxiliary Air and Circulating Pumps.

Diameter of steam cylinder, inches	6
circulating-pump cylinder, inches	8
air-pump cylinder, inches	8
Stroke of pistons, inches	7
Diameter of circulating-pump suction, inches	4
discharge, inches	4
air-pump suction, inches	4
discharge, inches	3
steam pipe, inch	1
exhaust pipe, inches	1¼

TURNING GEAR.

There is a worm wheel on the main shaft at the after end of the main H.P. and each L.P. turbine, engaging with a removable worm and shaft, turning being accomplished by means of a ratchet wrench applied to the worm shaft.

FORCED-DRAFT BLOWERS.

The forced draft is of the closed fireroom type, there being two blowers in each fireroom, driven by direct-connected enclosed two-cylinder horizontal steam engines. The shafts of the blowers are vertical, the blowers being close up under the deck with the engines below them. Lubrication is accomplished by a pump driven by an eccentric on the engine shaft, oil passages and tubes being fitted to the various journals.

Data for One Blower.

	As finally installed
Type of fan	Sturtevant.
Diameter of fan, inches	54
Width at tip of vane, inches	10½
Number of vanes	9
Revolutions per minute, designed	850
Discharge pressure, designed, inches of water	5
Diameter of engine cylinders, inches	4½
Stroke of pistons, inches	5

LUBRICATING SYSTEM.

Oil for all the turbine bearings and the main-shaft steady bearings is supplied by a forced-lubrication system working under a pressure of about 35 pounds per square inch. There are two 4½-inch by 5-inch by 6-inch Blake vertical simplex steam pumps which pump the oil from two 635-gallon drain tanks through a cooler to the various bearings. The oil from the bearings returns by gravity to the drain tanks, which are situated low down in the after end of the engine room. Sight glasses are fitted on the return pipes at all the bearings so that the flow of oil can be observed. Thermometers are fitted in the pipes to and from the cooler and in the drain pipes from the forward bearings of the main H.P. and cruising turbines and from forward and after bearings of the L.P. turbines.

There is a 4½-inch by 5-inch by 6-inch Blake vertical simplex steam pump for circulating water through the oil cooler. This pump draws from the auxiliary sea valve and discharges through the cooler into the main outboard-delivery pipe. There is also a connection from the fire main to supply circulating water if the circulating pump should be disabled.

The cooler is cylindrical in shape and has a sheet-copper shell and brass tubes and heads. The tubes are packed at the tube sheets with cotton-tape packing and screwed ferrules. The oil flows through the tubes and the circulating water through the shell around the tubes. By means of baffles in the shell and heads both oil and water are made to travel the length of the shell four times, the direction of flow of the oil being opposite to that of the water. After the preliminary trial, retarders were fitted in the tubes to increase the efficiency of the cooler.

Particulars of Oil Cooler.

Diameter of shell, inside, inches	0–18⅝
Thickness of shell, U. S. S. G., No	10
Length between tube sheets, feet and inches	4–02
Thickness of tube sheets, inch	0–00¾

Number of tubes	222
Diameter of tubes, outside, inch	0–00$\frac{5}{8}$
Thickness of tubes, B.W.G., No	18
Cooling surface, square feet	151.8

There is a galvanized-steel oil-stowage tank in the engine room. This tank is divided into three compartments of the following capacities: lamp oil, 44 gallons; auxiliary lubricating oil, 80 gallons; turbine oil, 112 gallons. Pipes are provided for filling the tanks from the deck, and there is a pipe from the bottom of the turbine oil compartment to the oil-drain tanks for the purpose of supplying oil to, or making up leakage of the forced-lubrication system.

WATER SERVICE.

Water service is provided for the stern tubes by pipes taking water from the fire main. There is no water service to the main bearings or steady bearings, the forced-lubrication system being considered capable of preventing these bearings becoming dangerously hot.

HOISTING GEAR.

A system of trolleys and chain hoists is installed for lifting and moving aside the turbine cylinders and rotors, and also for the main condenser heads.

REVOLUTION COUNTER GEAR.

There is a continuous rotary-motion revolution counter for each shaft, the three counters being combined in one rectangular case at the working platform. The counters register one for every ten revolutions of the turbines, the unit revolutions being read from graduations on the first numbered wheel of each counter or from a graduated numbered disc on each counter-driving shaft; the latter also serve as motion indicators for the turbines. The counters are driven by shafting, with bevel gears and universal joints, from worms on the forward ends of the turbine shafts.

EVAPORATING AND DISTILLING PLANT.

There are two evaporators and two distillers, the rated capacity of the plant being 1,000 gallons of fresh water per 24 hours. The evaporators are in the engine room at the forward end, and the distillers are on the forward end of the engine-hatch coaming on the main deck. The evaporator shells are vertical cylinders, built up of steel plates and angles. The steam coil of each evaporator consists of forty seamless-drawn brass tubes, bent U shape, and a brass steam head with steel-plate cover.

Particulars of One Evaporator.

Diameter of shell, inside, feet and inches	2-04
Length of shell, inside, feet and inches	5-02⅝
Thickness of shell, inch	00 3/16
Number of tubes	40
Diameter of tubes, outside, inches	01½
Thickness of tubes, B.W.G., No.	13
Heating surface, square feet	68
Working pressure, shell, pounds per square inch	30
coil, pounds per square inch	85
Diameter of vapor outlet, inches	02½
feed inlet, inch	00¾
steam inlet to coil, inch	01
drain from coil, inch	00½

The distillers are cylindrical in shape, the shells being of sheet copper; the heads, tubes and tube sheets of brass. The tubes are expanded into both tube sheets, and expansion in the direction of their length is provided for by a cylindrical extension of the bottom tube sheet being made to fit a stuffing box formed by the bottom head. One distiller was originally installed in a horizontal position under the main deck directly over the evaporators. The tests of the plant showed that with distiller in this position, it required constant attention to prevent priming of the evaporators, and that the distilled water was exceedingly warm. The distiller was, therefore, moved to its present position on the engine-hatch coaming, and another distiller of the same dimensions was added. In the new installation both distillers are in a vertical position.

Particulars of Distillers.

Diameter of shell, outside, inches	0–09
Length of shell, feet and inches	2–05¼
Thickness of shell, inch	0–00 7/16
Number of tubes	43
Diameter of tubes, outside, inch	0–00⅝
Thickness of tubes, B.W.G., No.	17
tube sheets, inch	0–00½
Cooling surface (one distiller), square feet	17.0
Diameter of vapor inlet, inches	0–02
fresh-water outlet, inch	0–00¾
circulating-water inlet and outlet, inches	0–01¼

There is a 3½-inch by 4-inch by 4-inch evaporator-feed pump and a 2¾-inch by 3-inch by 4-inch fresh-water pump, both of Blake vertical simplex type. The distiller circulating water is supplied by the engine room fire and bilge pump.

HEATING SYSTEM.

The steam-heating system is in two sections, one forward and one aft. The forward section consists of three circuits, one for crew's quarters, pilot house, etc., one for officers' quarters, and one for galley and pantry. Steam is supplied by a 1-inch branch, with reducing valve, from the auxiliary steam pipe in the forward fireroom. The after section consists of one circuit leading to all the living quarters in the after part of the vessel. Steam is supplied by a 1-inch branch, with reducing valve, from the auxiliary steam pipe in the engine room. The drains from the forward circuits are led to a 1-inch trap in the forward fireroom, and those from the after circuit to a 1-inch trap in the engine room, the discharge from both traps being led to the feed tank and condensers.

AIR COMPRESSOR PLANT FOR TORPEDOES.

In the after end of the engine room there is a Mahn Navy-standard type air compressor for supplying compressed air to the torpedo tubes. The discharge from the compressor passes through a combined separator and cooler to an accu-

mulator from which pipes are led to the torpedo tubes. Air compression is accomplished in three stages, each stage having two cylinders, all single acting. Motive power is furnished by two steam cylinders. All the cylinders are vertical, the piston rods being all attached to one crosshead, from which there is a connecting rod driving a crank shaft equipped with two flywheels. Circulating water for the jacket and cooler is supplied by two pumps driven from the crank shaft.

Particulars of Air Compressor.

Diameter of low-pressure air cylinders, inches	4.64
intermediate-pressure cylinders, inches	1.97
high-pressure cylinders, inch	0.787
steam cylinders, inches	5.75
Stroke of all pistons, inches	5.04
Revolutions per minute, designed	500
Steam pressure, designed, pounds per square inch	75–150
Rated capacity in cubic feet of air delivered per hour, at 2,500 pounds per square inch	15

There is an air main running through the engine and fire-rooms with connections for driving pneumatic tools and for blowing soot off the boiler tubes. This main is supplied by a branch from the air-compressor discharge through a reducing valve which reduces the pressure to 100 pounds.

ANCHOR GEAR.

The capstan is on the forecastle deck and consists of a gypsy head and wildcat on a vertical shaft driven through worm gearing by a 4-inch by 6-inch two-cylinder vertical steam engine, which is located on the main deck directly under the capstan.

STEERING GEAR.

In the pilot house there is a 5-inch by 5-inch vertical two-cylinder steam-steering engine connected to the rudder quadrant by $\frac{3}{4}$-inch chains and $\frac{3}{8}$-inch steel-wire ropes, leading along the main deck close to the gunwales. There is a con-

nection to the bridge so that the vessel can be steered by steam from either pilot house or bridge. There is a pair of hand-steering wheels on the main deck aft, and emergency hand steering is provided for by relieving tackles having direct connection to the rudder quadrant.

ELECTRIC GENERATORS.

On a platform in the after end of the engine room there is a 5-kilowatt General Electric Company dynamo, driven by a horizontal Curtis steam turbine.

TORSION METERS.

The torsion meters used on these vessels are of the contractors' own make, from designs of Mr. J. F. Metten, Chief Engineer of the Cramp Company. They are self-contained and their action is entirely mechanical, no electric current or other outside agency being required for their operation, and no special skill is required in taking records by them.

The following is Mr. Metten's description of these torsion meters and the method used in calibrating same, reference being had to Plates III, IV, and V:

This apparatus is designed for measuring the torsional deflection of rotating shafts so that the power being transmitted can be readily ascertained when the speed of rotation is known.

All torsion meters for measuring shaft powers are based on the same principle, that ordinary materials are perfectly elastic when stresses do not approach close to the elastic limit of the material. The two drawings, Plates III and IV, show the details of the torsion meters and drawing, Plate V, shows the application of the device to a turbine shaft on a vessel with diagrams illustrating clearly the method of measuring power being transmitted. Referring to Plate III, *A* is a sleeve rigidly attached to the shaft at *X* and loosely resting on shaft at *Y*, which is shown in halves for convenience in installing on shafts with solid couplings. The extension arms at the end *Y* are for operating the measuring pointers, as will be described later. Only one of the latter is used, the other being cast on simply for balancing.

B is a disc securely fixed to the shaft, also made in halves. This disc is located close to the sleeve at *Y* as shown in large scale section, Fig. 1.

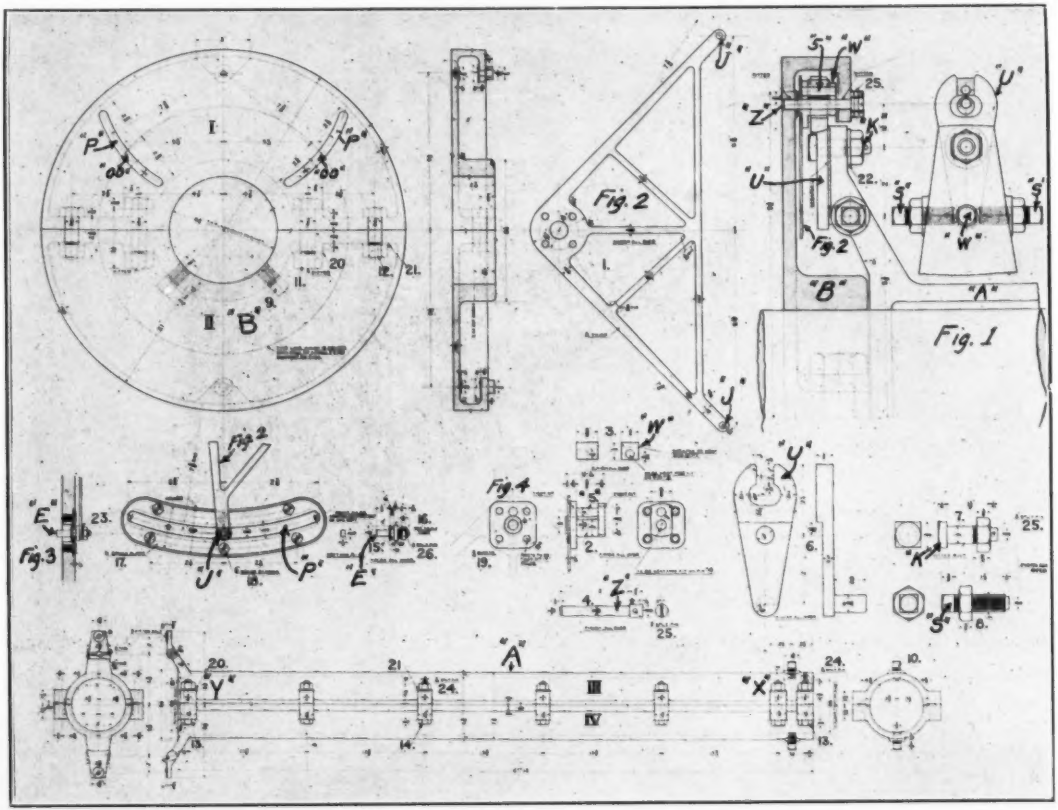


Plate III.—DETAILS OF TORSION METER.

Pivoted on this disc at Z is the indicator hand, Fig 2, which is riveted to a crank forging, Fig. 4, the indicator and crank oscillating on pin Z. This crank is provided with a pin 5 on which is mounted a block W the sides of which are a sliding fit in fork U. This fork is attached to one

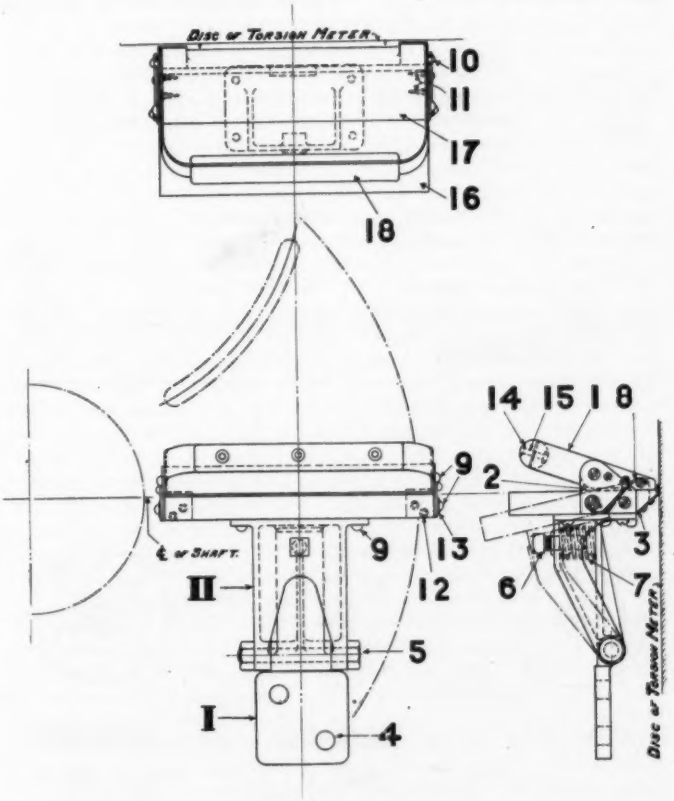


Plate IV.

CARD TABLE FOR TORSION METER.

of the extension arms on sleeve A by means of a fitted bolt K, and the projecting lug on end of fork extends into an opening drilled in the center of extension arm as shown, the opening being larger than the lug, which is held between the ends of two set screws S, the object of which will be explained further on. It will be apparent that any twisting of

the shaft due to its elastic properties between the fixed end of sleeve *A* at *X* and the fixed disc at other end of sleeve will cause the fork end of *U* to move relative to disc, the direction and magnitude of the movement depending on the direction and magnitude of twisting load on the shaft. This movement of the fork which engages the pin 5 through the block *W* causes the indicator hand, Fig. 2, to move through an arc the extent of

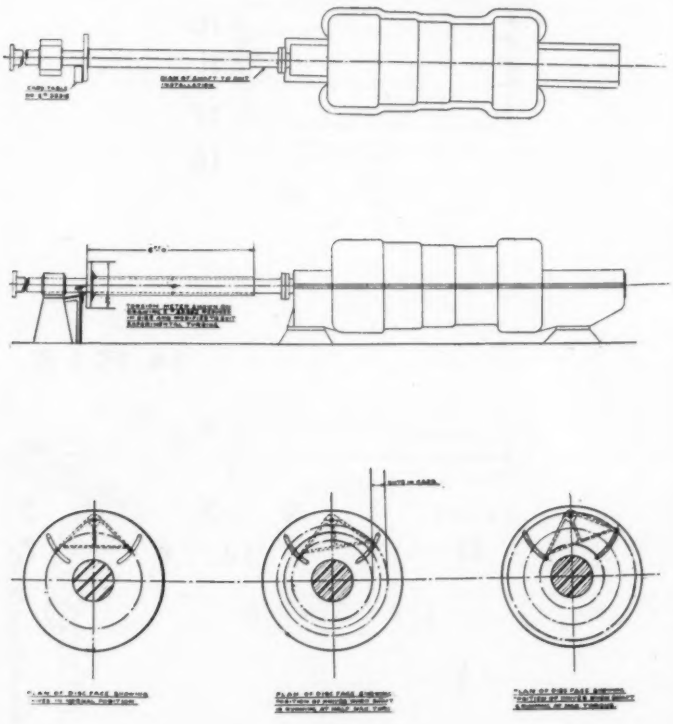


Plate V.

ARRANGEMENT OF TORSION METER ON SHAFT.

which is dependent upon the relative movement of the fork arm mentioned above. The indicator hand is provided with two knives *B* at the two extreme points *J* which extend through curved slots *P* in the disc *B*. These knives are made with shoulders that slide on a brass plate fastened to back of disc as shown in Fig. 3, the object being to provide for free movement along the arc of travel, but to prevent motion in a line parallel

with axis of shaft. These knives are arranged to project about ¼ inch through face of disc, and when the apparatus is assembled complete and the shaft under no torsional strain, the points at each end of indicator hand, *JJ*, stand at the position *OO* on disc *B*. The two set screws *SS* engaging lug on form arm *U* are for effecting this adjustment, as will be understood clearly from the drawing.

When the knives stand at the position *OO* and the shaft and instrument is revolved, but under no torsional strain, it will be noted that both knife points revolve at exactly the same radius, and if the edge of a sheet of paper is held against the face of the disc both knives will cut the edge at the same point or, in other words, only one cut will be made. If the shaft is under a torsional strain while being revolved the movement of the indicator hand causes one of the knives to move from the normal line towards the periphery of disc and the other to move towards the center of shaft, the amount of which is proportional to the twisting force exerted on the shaft. Thus if the instrument and shaft is rotated and the shaft is under a torsional strain, the two knives will revolve at different radii, the actual difference between which will be twice the movement of the knife, as one moves towards the center and one away from the center of disc. If the edge of paper is held against face of disc when shaft and instrument is revolving under latter conditions there will be two cuts in the edge of the paper, the distance which cuts are separated representing the difference in radii at which the knives are revolving; this difference being affected directly and only by the torsional deflecting of the shaft, its magnitude will vary directly with the magnitude of the torsional load on the shaft and the two cuts on the paper will be a permanent record of the twisting force on the shaft at the time the cuts were made. In order to compute the power transmitted it is only necessary to know the speed of revolutions at which record is taken and the twisting moment represented by the distance the cuts are separated. The latter is determined as follows:

The shaft upon which the instrument is to be fixed is mounted on suitable bearings, preferably of roller type to eliminate friction, and between the bearings the torsion meter is installed and rigidly attached to the shaft as described above with the two pointers adjusted so as to both come exactly on the normal radius. One end of the shaft is firmly fixed to prevent its turning and at the other end is attached a lever of known length, and by means of suitable scales or known weights twisting loads of known magnitude are applied to the shaft varying from small amounts up to the extreme loads the shaft is intended to be subjected to in service. During the application of each load to the shaft the distance the pointers or knives have moved from the normal line is carefully measured and a scale is prepared showing the twisting moments in inch or foot pounds represented by movements of the pointers.

Plate IV shows the small table which is installed when torsion meter is in its final position on its shaft in the ship. This is a small wooden table with the edge parallel with the face of disc *B* and provided with a

clamp for holding the record cards in proper position. This table is hinged at its base, which is provided with stops and springs arranged so as to keep the table away from the revolving disc and knives ordinarily, but which after a card has been inserted allows it to move back to a position where the revolving knives will cut the edge of the paper. The stop is adjustable so as to prevent the table touching the disc. The base of the table is rigidly attached to any convenient fixed part of the ship's structure. The arrangement shown is adapted particularly for shafts revolving at comparatively high speeds, and ordinarily the record cuts will be made as quickly as the table can be operated. The apparatus shown is for shafts transmitting uniform torque. The advantages claimed over other types of torsion meters are as follows:

The records are permanent and can be referred to at any time and do not depend on the accuracy of the operator. The power can be taken with extreme accuracy, as there are few joints, and these can be given ample wearing surface.

The arrangement of two knives or measuring points on opposite sides of the fulcrum gives double the length of scale that could be obtained by ordinary means with the same centers. This allows actuating points to be given sufficient diameter and distance from fulcrum to avoid the difficulties experienced with ordinary devices for multiplying deflection on the scales. The instrument being calibrated directly with its shaft, errors of observation, etc., are eliminated. The records themselves are of the best type for accurate measurements, being clean sharp cuts about 3/16 inch in depth on the edge of a substantial paper card. The moving parts of the recording mechanism are so arranged as to permit of being dynamically balanced, which is very important at high speeds to insure accuracy.

U. S. S. "SMITH."

Length of shaft calibrated, feet and inches	6-0
Distance between cuts, in inches, for each 1,000 pounds, applied at 6 feet leverage. — Center shaft	1.1111
Port shaft	1.1711
Starboard shaft	1.08444
Horsepower constant, center shaft	1.0281
port shaft	.9755
starboard shaft	1.0535

Shaft horsepower $= C \times R \times F$.

C = Constant as given above.
R = Revolutions of shaft per minute.
F = Distance between cuts, in inches.

Deduction of formulae:

L = Length arm, in feet.
P = Pull, in pounds.
R = R.p.m.

$$\text{H.P.} = \frac{2\pi LPR}{33{,}000}.$$

$$\frac{\text{H.P.}}{R} = \left(\frac{2\pi L}{33{,}000}\right) P = .0011424\,P.$$

With $P = 1{,}000$ pounds, distance between cuts = 1.111 inches for center shaft

$$\therefore \text{For center shaft } \frac{\text{H.P.}}{R} = \left(.0011424 \times \frac{1{,}000}{1.111}\right) \times F.$$

$$= C \times F. \qquad C \text{ for center shaft} = 1.0281$$

port shaft = .9755

starboard shaft = 1.0535

$$\text{or H.P.} = C \times F \times R.$$

Weights of Machinery.

	Smith. Pounds.	*Lamson.* Pounds.
Main turbines	97,550	98,056
Shafting	27,025	26,840
Main shaft bearings	3,477	3,473
condensers	32,856	32,834
air and circulating pumps	11,384	11,457
Propellers	3,194	3,188
Boilers	132,446	133,576
Boiler fittings	35,103	34,662
Smoke pipes and uptakes	7,934	8,013
Steam and exhaust pipes and valves	30,700	31,653
Suction and discharge pipes and valves	21,490	22,097
Lagging and clothing	20,648	20,648
Flooring and gratings, etc	13,223	12,936
Auxiliaries	24,520	24,461
Fittings and gear	12,515	12,509
Stores, tools and spare parts	9,450	9,445
Miscellaneous machinery, etc	14,375	14,280
Steam and exhaust connections, outside machinery compartments	1,916	1,900
Total, without water	499,806	502,028
Water	59,741	59,733
Total, with water	559,547	561,761
= Tons (2,240 pounds)	249.80	250.79
	Tons.	Tons.
Penalty weight	275.00	275.00
Actual weight	249.80	250.79
Difference	25.20	24.21

The following is a comparison of the machinery weights of the *Smith* with those of the U. S. T. B. destroyer *Bainbridge*, the latter being equipped with twin-screw reciprocating engines and Thornycroft boilers. As no official record of horsepower was taken on the full-power trial of the *Bainbridge*, the designed I.H.P. (8,000) of that vessel and the designed equivalent I.H.P. (10,000) of the *Smith* are used for purposes of

comparison. The weight of water under full-power steaming conditions is included in all weights used.

	Smith.	*Bainbridge.*
Engine-room weights, pounds per I.H.P	25.71	25.84
Propellers and all shafting and bearings outside of engine rooms, pounds per I.H.P	2.79	3.97
Fireroom weights, pounds per I.H.P	26.72	28.22
Total weight of machinery, pounds per I.H.P	55.95	58.47

It will be noted that there is very little saving in engine-room weights by the use of turbines, this probably being due to the fact that cruising turbines are required with this type of machinery. There is a material saving (1.18 pounds per I.H.P.) in the weight of propellers and shafting in the case of the turbine installation. The difference in fireroom weights is probably due to the different boilers used, and can hardly be credited to the use of turbines. The total machinery weight per I.H.P. of the *Smith* is 2.52 pounds less than that of the *Bainbridge*.

BUILDERS TRIALS.

Smith.—Dock trials of the machinery of the *Smith* were run May 11th to 16th, 1909. The main and auxiliary machinery was operated under various conditions, and observations of the performance were made by the builders in preparation for their own and the official sea trials. Builders' sea trials were conducted in the vicinity of Delaware Breakwater, May 17 to 24, 1909.

The performance of the propelling engines was very satisfactory, the contract speed of 28 knots being obtained with about 700 revolutions per minute. As this was somewhat below the rate at which the turbines were designed to revolve, the builders reduced the area of the propellers after these trials by taking one inch off the tips of all blades.

It was found necessary to run the main air pumps at a speed reaching at times 90 double strokes per minute, a much higher speed than that for which they were designed, but this was probably due to the turbine drains being kept wide open much of the time. This condition was remedied on the official trials.

These pumps were run at about 65 double strokes per minute on the four-hour full-speed trial.

The oil cooler for the lubricating system did not prove entirely efficient, but this was remedied by fitting retarders in the oil-cooler tubes.

The boilers worked very efficiently, no trouble being experienced with them except that furnace-door linings burned out very rapidly.

Lamson.—Dock trials of the machinery of the *Lamson* were made on October 19, 20 and 21, 1909. The main and auxiliary machinery was run under various conditions and everything worked satisfactorily. While the vessel was maneuvering out of her slip on November 4, 1909, the port L.P. turbine became disabled. Examination of the turbine showed that about twelve rows of blading in the rotor and about ten rows in the casing were impaired to such an extent as to require renewal. The damaged blades were in the third, fourth and fifth expansion stages. The cause of this damage was not definitely determined, but was probably due to distortion of the rotor by unequal heating due to steam from the gland-system relief valve entering the turbine before it was warmed up. The relief valve from the gland system discharges into the L.P. turbine casing at the fifth expansion. The damaged blading having been renewed, a builders' trial was made on the Delaware Bay on November 23, 1909, during which all the parts of the machinery installation worked satisfactorily.

OFFICIAL TRIALS.

STANDARDIZATION TRIALS.

Smith.—On June 22, 1909, standardization trials were held consisting of fifteen runs over the Delaware Breakwater measured-mile course at the mouth of Delaware Bay. One of the blowers being out of service, the highest speed for one run was 26.924 knots.

An air trunk was fitted, communicating the two firerooms and with three blowers supplying the air for both firerooms, five more standardization runs were made on June 25, 1909,

the speed for the fastest run being 28.431 knots. On September 22, 1909, new blowers having been installed, five high-speed standardization runs were made, the speed for the fastest run being 31.901 knots.

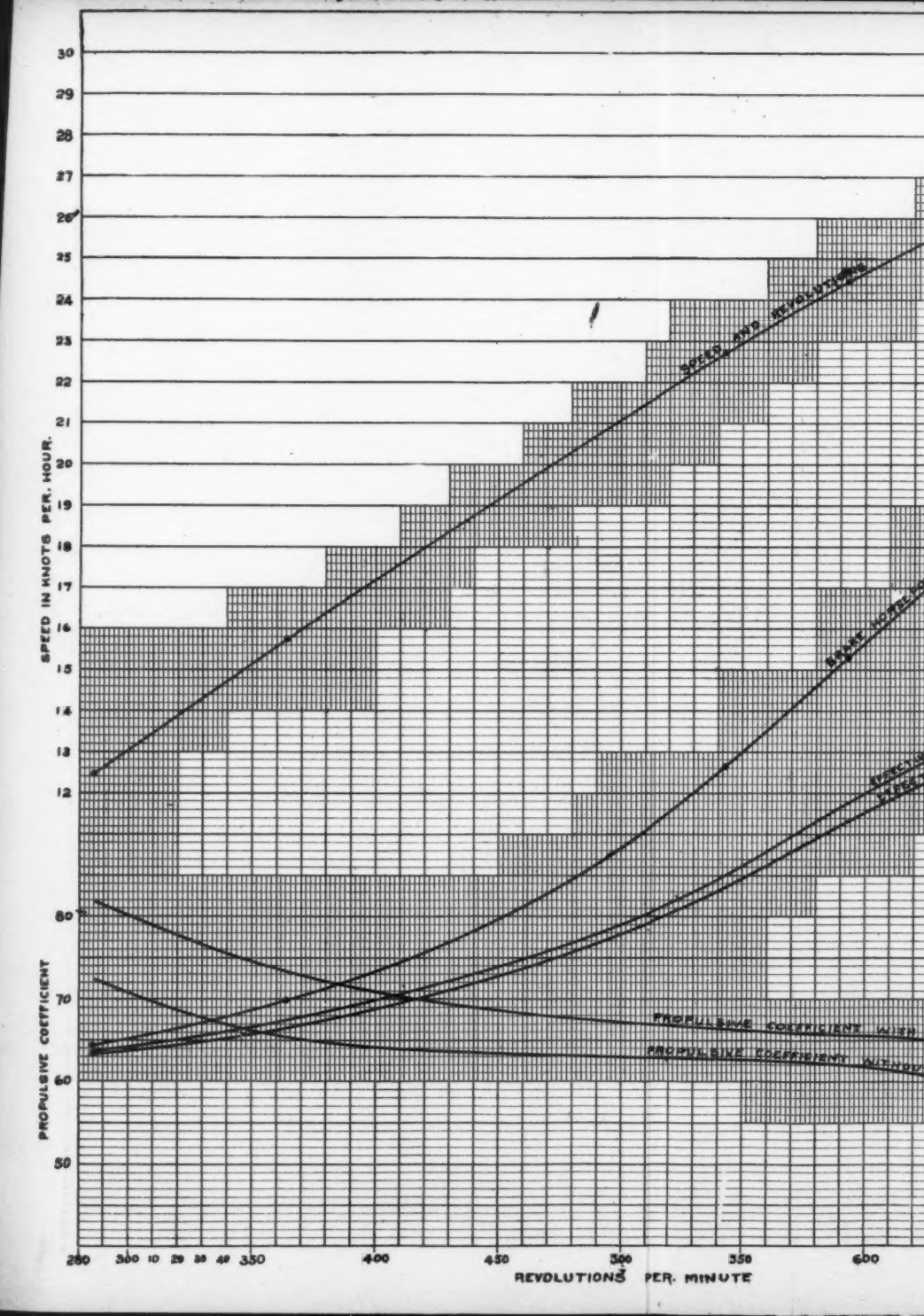
Curves of speed, revolutions and horsepower, with propulsive coefficients, deduced for the runs on June 22, 1909, are given on Plate VI.

Lamson.—On December 2, 1909, standardization trials were held consisting of fourteen runs over the Delaware Breakwater measured-mile course, and on December 3, 1909, eight more runs were made over the course. Curves plotted with the data obtained on these runs are given on Plate VII.

DATA OF STANDARDIZATION TRIALS.

Smith, June 22, 1909; displacement, 677 tons.

Number of run.	Mean R.P.M., three shafts.	Total S.H.P., three shafts.	Speed, knots.
1	638.269	6,415	24.913
2	644.777	6,525	26.924
3	632.128	6,190	24.734
4	542.005	3,765	23.672
5	541.549	3,834	21.539
6	547.394	3,884	24.081
7	594.387	5,208	23.242
8	595.987	5,190	25.510
9	588.664	5,004	23.425
10	359.907	933	16.129
11	361.269	989	15.254
12	375.132	1,084	16.365
13	284.867	414	13.061
14	290.190	441	11.786
15	282.201	416	13.458
June 25, 1909; displacement, 710 tons.			
16	730.462	9,966	28.251
17	702.878	8,926	27.986
18	709.810	9,316	28.022
19	725.166	9,835	28.431
20	715.377	9,165	28.251
September 22, 1909; displacement, 714 tons.			
21	811.25	12,039	31.289
22	805.00	11,824	28.669
23	786.52	11,636	30.887
24	774.64	11,471	27.881
25	826.38	11,603	31.901


SPEED IN KNOTS PER HOUR.
30
29
28
27
26
25
24
23
22
21
20
19
18
17
16
15
14
13
12
PROPULSIVE COEFFICIENT
80
70
60
50
PROPULSIVE COEFFICIENT WITH
PROPULSIVE COEFFICIENT WITHOUT
250
300
10
20
30
40
350
400
450
500
550
600
REVOLUTIONS PER MINUTE

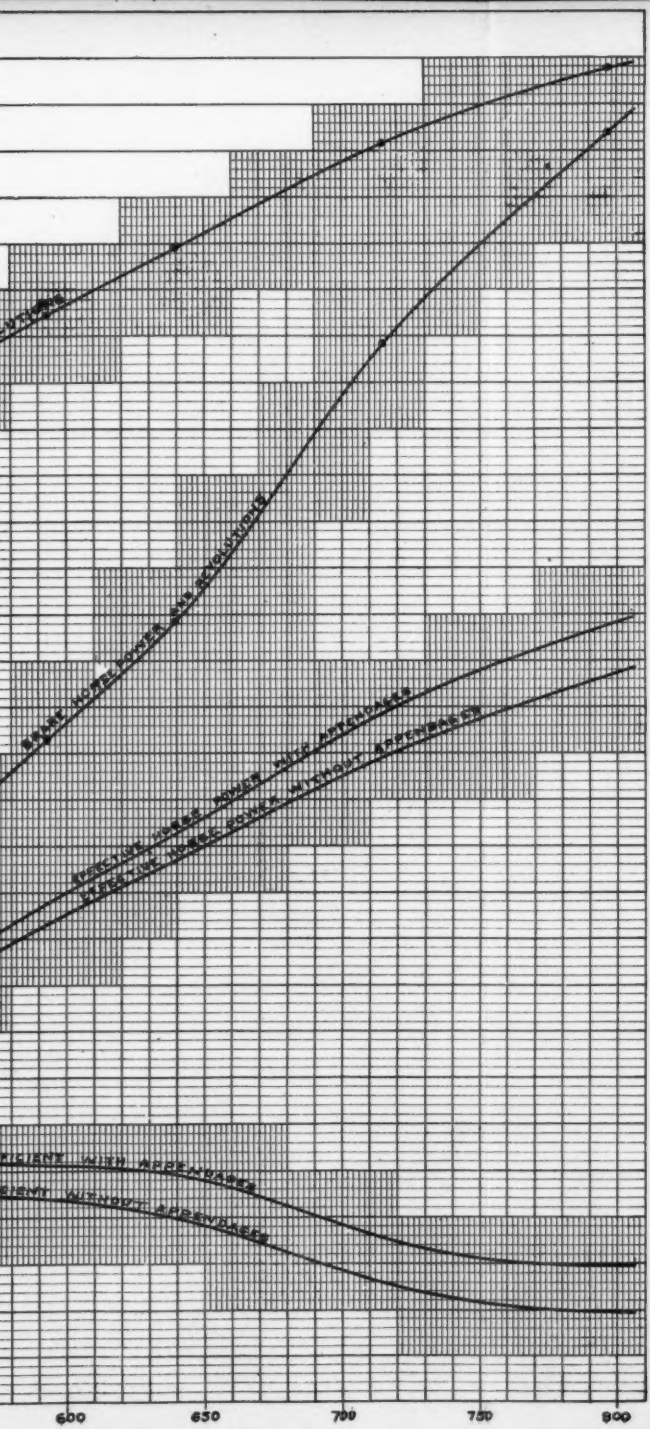


Plate VI.

U.S.S.SMITH.

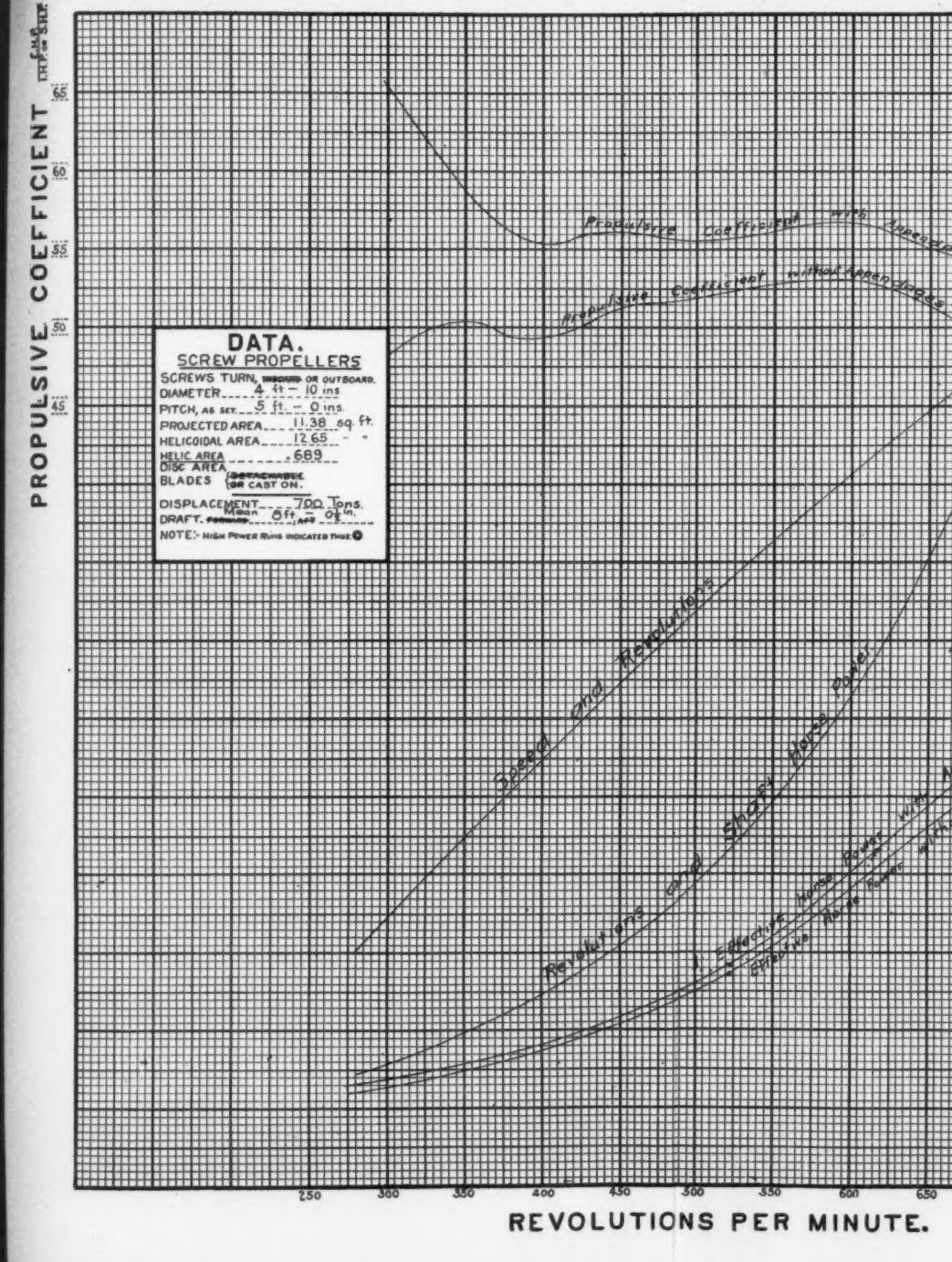
STANDARDIZATION RUNS

DELAWARE BREAKWATER

JUNE 22-1909.

PROPELLERS

NUMBER of PROPELLERS	3
" " BLADES EACH	3
DIA.	4'-10"
PITCH AS SET (MEAN)	5'-0"
PROJ. AREA	11.87 □'
DEL. AREA	13.19 □'


PROPULSIVE COEFFICIENT
65
60
55
50
45
DATA.
SCREW PROPELLERS
SCREWS TURN, INBOARD OR OUTBOARD.
DIAMETER 4 ft – 10 ins
PITCH, AS SET. 5 ft. – 0 ins.
PROJECTED AREA 11.38 sq ft.
HELICOIDAL AREA 12.65 "
HELIC AREA .689
DISC AREA
BLADES (DETACHABLE. OR CAST ON.
DISPLACEMENT 700 Tons.
DRAFT. Mean 6ft – 0in.
NOTE:- HIGH POWER RUNS INDICATED THUS ⊙
Propulsive Coefficient with Appendages
Propulsive Coefficient without Appendages
Speed and Revolutions
Shaft Horse Power
Revolutions and Shaft Horse Power
Effective Horse Power
250
300
350
400
450
500
550
600
650
REVOLUTIONS PER MINUTE.

INDICATED, SHAFT AND EFFECTIVE HORSEPOWER.
SPEED IN KNOTS PER HOUR.
12000
11000
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
0
32
30
28
26
24
22
20
18
16
14
12
10
U. S. S. LAMSON
TRIAL STANDARDIZATION
COURSE DELAWARE BREAKWATER
DATE DECEMBER 2nd and 3rd 1902
BUREAU OF STEAM ENGINEERING, NAVY DEPARTMENT
WASHINGTON, D. C.
REMARKS
LETTER No. 23604-D.
FORM No. 113.
600
650
700
750
800
850
INUTE.

Plate VII.

Lamson, December 2, 1909; displacement, 694 tons.

Number of run.	Mean R.P.M. three shafts.	Total S.H.P., three shafts.	Speed, knots.
1	291.92	502	12.184
2	288.35	481	13.082
3	286.07	487	11.274
4	356.54	1,016	16.342
5	359.71	1,030	14.024
6	362.68	1,073	16.866
7	730.96	10,274	26.826
8	778.39	11,954	30.923
9	751.38	11,276	27.225
10	793.93	11,727	31.407
11	818.76	12,660	28.377
12	799.73	12,197	31.408
13	774.66	9,830	27.677
14	785.85	12,014	31.090
December 3, 1909; displacement, 687 tons.			
15	564.44	4,345	22.546
16	567.39	4,295	23.548
17	571.43	4,401	23.101
18	704.80	9,207	27.082
19	702.58	9,375	27.589
20	708.98	9,230	27.606
21	702.18	8,935	27.669
22	714.03	9,357	27.697

FOUR-HOUR FULL SPEED TRIALS.

Smith.—An attempt was made to run a four-hour full-speed trial on June 26, 1909, with three blowers supplying the air for both firerooms. After running about two hours and three quarters, this trial was stopped on account of the break-down of one of the blowers. After new blowers had been installed, a successful four-hour trial was held on September 23, 1909. The trial was begun at 8:45 A. M., off the entrance to Delaware Bay and ended four hours later off Fenwick's Island light ship. The weather was clear with light breezes from S.S.W., and the sea was smooth with gentle swell from the S.E. Average displacement for trial, 716 tons.

Lamson.—The four-hour full-speed trial was run off the entrance to Delaware Bay on December 4, 1909, beginning at 10:15 A. M., with moderate breezes from W.N.W. and moderate sea. Average displacement for trial, 690 tons.

PERFORMANCE.

Four-Hours, Full-Speed, Trials.

Pressures:	*Smith.*	*Lamson.*
Average steam pressure at boilers, gauge, pounds	242.0	247.0
engines, gauge, pounds	239.0	246.0
main H.P. turbines, steam belt, absolute, pounds	199.0	204.0
L.P. turbine, steam belt, starboard, absolute, pounds	45.6	47.9
L.P. turbine, steam belt, port, absolute, pounds	45.6	46.4
on turbine glands, gauge, lbs.	2.6	0.7
pressure in oil system, gauge, pounds	5.7	5.1
Vacuum in condenser, starboard, inches of mercury	25.9	26.8
port, inches of mercury	25.8	27.4
Revolutions, or double strokes, per minute:		
Shaft No. 1, starboard	724.69	755 80
2, center	728.65	728.96
3, port	717.99	751.71
Average	723.78	745.48
Pumps, main air, starboard	65.9	75.0
port	65.9	73.0
circulating, starboard	309.9	327.0
port	305.2	258.0
feed, forward	32.9	39.0
after	31.6	38.0
oil, No. 1	58.9	13.0
2	52.3	24.0
cooler circulating	78.8	38.0
fire and bilge	38.4	49.0
Forced-draft blowers (average of four)	869.0	789.0
Speed of ship, knots	28.35	28.605
Slip of propeller, in per cent. of its own speed, based on mean pitch:		
Shaft No. 1	20.72	23.33
2	21.15	20.51
3	19.98	22.91
mean	20.616	22.25
Air pressure in firerooms, inches of water	5.0	4.2
Shaft horsepower:		
Shaft No. 1, starboard	3,303.0	3,543.0
2, center	3,755.0	3,626.0
3, port	2,888.0	3,428.0
Total	9,946.0	10,597.0

	Smith.	*Lamson.*
Indicated horsepower:		
Air, circulating and feed pumps	149.0	172.0
All other auxiliaries in operation	267.0	272.0
Total auxiliaries in operation	416.0	444.0
Collective horsepower, main engines and auxiliaries, S.H.P. and I.H.P.	10,362.0	11,041.0
Temperatures:		
Injection, degrees F.	69.6	54.0
Discharge, degrees F.	107.0	83.0
Feed to heater, degrees F.	107.0	96.0
Feed from heater, degrees F.	187.0	165.0
Outside air, degrees F.	79.1	...
Engine room, working level, degrees F.	126.0	82.0
Smoke-pipe gases, degrees F.	572.1	622.6
Coal:		
Kind and quality	Pocahontas, hand-picked.	Pocahontas, excellent.
Pounds used per hour	16,800.0	20,834.0
Knots run per ton of coal	3.8	3.141
Water:		
Pounds per hour	195,190.0	162,173.0
Pounds per H.P. per hour	18.837	14.688
(H.P. = S.H.P. of main engines plus I.H.P. of auxiliaries.)		
Deduced data:		
S.H.P. per square foot of grate surface	26.99	28.76
heating surface	.552	.5886
Pounds of coal per S.H.P. per hour	1.689	1.923
square foot grate surface per hour	45.59	55.23
Cooling surface (main condensers) square feet per S.H.P.	.807	.757

TWELVE-HOUR TRIALS AT TWENTY-FOUR KNOTS.

Smith.—The twelve-hour endurance and coal-consumption trial at twenty-four knots was begun off Cape Henlopen at 7:25 A. M. on June 23, 1909, and ended twelve hours later off Overfalls light ship. The weather was fair with gentle to moderate breezes from W. by S. to S.W., and the sea smooth. Average displacement for trial, 697 tons.

Lamson.—The twelve-hour endurance and coal-consumption trial at twenty-four knots was run on December 5, 1909, off the Delaware Capes, with gentle breezes from N.N.W. and smooth sea. Average displacement for trial, 693 tons.

Performance.—Twelve-Hours, Twenty-four-knot, Trials.

	Smith.	*Lamson.*
Pressures:		
Average steam pressure at boilers, gauge, pounds	251.0	247.0
engines, gauge, pounds	251.0	247.0
I.P. cruising turbine, steam belt, absolute, pounds	249.0	249.0
main H.P. turbine, steam belt, absolute, pounds	95.0	109.0
L.P. turbine, steam belt, starboard, abs., pounds	22.5	26.5
L.P. turbine, steam belt, port, absolute, pounds	22.0	26.3
on turbine glands, gauge, lbs.	2.5	1.2
Pressure in oil system, gauge, pounds	6.5	6.0
Vacuum in condenser, starboard, inches of mercury	27.0	27.9
port, inches of mercury	27.0	28.8
Revolutions, or double strokes, per minute:		
Shaft No. 1	658.07	678.91
2	568.74	574.88
3	579.53	594.65
Average	602.11	616.14
Pumps, main air, starboard	54.0	55.0
port	57.0	55.8
main circulating, starboard	282.0	291.0
port	278.0	281.0
feed, forward	25.0	41.5
after	24.0	...
oil, No. 1	29.0	27.8
2	25.0	30.8
cooler circulating	76.0	33.4
fire and bilge	38.0	44.5
Forced-draft blowers (average of two)	555.0	569.0
Speed of ship, knots	24.696	24.804
Slip of propeller, in per cent. of its own speed, based on mean pitch:		
Shaft No. 1	23.94	25.96
2	12.0	12.56
3	13.64	15.47
Mean	16.526	17.996
Air pressure in firerooms, inches of water	1.07	1.07
Shaft horsepower:		
Shaft No. 1	2,863.0	...
2	1,486.0	...
3	1,227.0	...
Total	5,576.0	*5,740.0

*The torsion meters not working accurately; S.H.P. estimated by comparison with standardization trial and with the performance of the sister ship *Smith*.

	Smith.	*Lamson.*
Indicated horsepower:		
Air, circulating and feed pumps	112.2	79.0
All other auxiliaries in operation	211.0	205.0
Total auxiliaries in operation	323.2	284.0
Collective horsepower, main engines and auxiliaries, S.H.P. and I.H.P.	5,899.2	6,024.0
Temperatures:		
Injection, degrees F	68.0	49.3
Discharge, degrees F	88.0	64.8
Feed to heater, degrees F	81.0	74.6
from heater, degrees F	197.0	176.0
Oil before cooling, degrees F	108.0	...
after cooling, degrees F	73.0	...
Steam at I.P. cruising turbine, degrees F	400.0	...
H.P. cruising turbine, degrees F	198.0	...
main H.P. turbine, degrees F	328.0	...
Smoke-pipe gases, degrees F	581.8	555.1
Engine room, working level, degrees F	...	99.7
Coal:		
Kind and quality	Pocahontas, run of mine, excellent.	
Pounds used per hour	10,075.0	11,397.0
Knots run per ton of coal	5.49	4.875
Water:		
Pounds per hour	91,712.0	80,349.0
Pounds per H.P. per hour (H.P. = S.H.P. of main engines plus I.H.P. of auxiliaries.)	15.547	13.338
Deduced data:		
S.H.P. per square foot of grate surface	16.71	15.58
heating surface	.3097	.319
Pounds of coal per S.H.P. per hour	1.807	1.98
sq. ft. of grate surface per hour	30.20	30.93
Cooling surface (main condensers) square feet per S.H.P.	1.44	1.398

During the twelve-hour 24-knot trial the following auxiliaries were in operation:

Auxiliaries in connection with the operation of the main engines, including feed pumps.

Dynamo, as necessary to efficiently light the ship.

The evaporating and distilling plant in continuous operation.

Flushing system in continuous operation.

Forced-draft fans as required.

Steering engine, as required for steering.

TWELVE-HOUR TRIALS AT 16 KNOTS.

Smith.—The twelve-hour endurance and coal-consumption trial at 16 knots was begun off the Delaware Breakwater at 7:00 A. M. on June 24, 1909, and ended twelve hours later off Overfalls Light Ship. The weather was fair and warm, gentle breezes from the S.W., smooth sea. Average displacement for trial, 686 tons.

Lamson.—The twelve-hour endurance and coal-consumption trial at 16 knots was run on December 1, 1909, off the Delaware Capes, with stiff breezes from N. by W. and choppy sea. Average displacement for trial, 693 tons.

Performance.—Twelve-Hours, Sixteen-knot, Trials.

	Smith.	*Lamson.*
Pressures:		
Average steam pressure at boilers, gauge, pounds	256.0	243.0
engines, gauge, pounds	256.0	242.0
H.P. cruising turbine, absolute, pounds	187.0	202.0
I.P. cruising turbine, absolute, pounds	61.0	83.0
main H.P. turbine, abs., lbs.	24.9	30.0
L.P. turbine, starboard, absolute, pounds	8.2	10.0
L.P. turbine, port, abs., lbs.	8.4	9.9
on turbine glands, gauge, lbs.	3.6	1.0
Pressure in oil system, gauge, pounds	3.1	4.0
Vacuum in condenser, starboard, inches of mercury	26.7	27.0
port, inches of mercury	26.8	27.0
Revolutions, or double strokes, per minute:		
Shaft No. 1	386.10	399.1
2	336.05	332.7
3	396.11	388.8
Average	372.75	373.5
Pumps, main air, starboard	42.5	57.0
port	42.3	57.0
circulating, starboard	268.0	224.0
port	258.0	224.0
feed, forward	9.6	30.0
after	17.3	...
oil, No. 1	14.7	16.0
2	21.0	15.0
cooler circulating	54.5	35.0
fire and bilge	32.2	24.0
Forced-draft blowers (one)	404.0	398.0

	Smith.	*Lamson.*
Speed of ship, knots	16.098	15.86
Slip of propeller, in per cent. of its own speed, based on mean pitch :		
Shaft No. 1	15.54	19.41
2	2.92	3.33
3	17.64	17.28
Mean	12.03	13.34
Air pressure in firerooms, inch of water	0.56	0.8
Shaft horsepower :		
Shaft No. 1	533 0	...
2	243.0	...
3	371.0	...
Total	1,147.0	*1,159.0
Indicated horsepower :		
Air, circulating and feed pumps	71.9	79.0
All other auxiliaries in operation	114.7	114.0
Total auxiliaries in operation	186.6	193.0
Collective horsepower, main engines and auxiliaries, S.H.P. + I.H.P.	1,333.6	1,352.0
Temperatures :		
Injection, degrees F.	69.0	46.5
Discharge, degrees F.	76.5	55.0
Feed to heater, degrees F.	76.0	63.0
from heater, degrees F.	194.0	194.0
Oil before cooling, degrees F.	107.8	...
after cooling, degrees F.	76.0	...
Steam at H.P. cruising turbine, degrees F.	372.3	...
I.P. cruising turbine, degrees F.	302.0	...
main H.P. turbine, degrees F.	250.4	...
Smoke-pipe gases, degrees F.	552.5	577.7
Engine room, working level, degrees F.	...	90.0
Coal :		
Kind and quality	Pocahontas, run of mine, very good.	
Pounds used per hour	3,117.0	4,082.0
Knots run per ton of coal	11.57	8.709
Water :		
Pounds per hour	33,570.0	35,989.0
H.P. per hour	25.13	26.619
(H.P. = S.H.P. of main engines plus I.H.P. of auxiliaries.)		
Deduced data :		
S.H.P. per square foot of grate surface	6.875	6.290
heating surface	0.0637	0.129
Pounds of coal per S.H.P. per hour	2.718	3.522
square foot of grate surface per hour,	18.68	22.15
Cooling surface (main condensers), sq. ft. per S.H.P.	6.996	6.924

* The torsion meters not working accurately ; S.H.P. estimated by comparison with standardization trial and with performance of sister ship *Smith*.

During these trials the following auxiliaries were in operation:

Auxiliaries in connection with the operation of the main engines, including feed pumps.

Dynamo as required to efficiently light the ship.

The evaporating and distilling plant as necessary to supply "make-up" feed water and water for all other purposes.

Flushing system was in continuous operation.

Steering engine was used as required for steering.

POST-TRIAL EXAMINATIONS OF MACHINERY.

Smith.—On October 1, 1909, the machinery of the *Smith* was opened up and examined. Everything was found in good condition with the following exceptions:

The rings in the piston of one blower engine were frozen.

The brick work of furnaces required renewing in places, especially on front walls.

The liners at furnace openings and cheek pieces were considerably burnt.

The liners in the back connections were burnt through and warped in places.

The zincs in the boiler drums were considerably warped.

Lamson.—After the official trials had been completed the main and auxiliary machinery of the *Lamson* was opened up and examined and found generally in good condition, with the following exceptions:

Some of the blades of the main H.P. turbine had their ends damaged, indicating that contact had occurred. The damage was principally confined to the third expansion stage, and was not serious enough to require renewal of the blades.

One of the blower engines was seriously disabled, having been thrown out of service on the four-hour trial.

TEST OF ANCHOR GEAR, SMITH.

On this test an anchor was let go in 27 fathoms of water and chain veered to 75 fathoms at the windlass. Total number of fathoms hove up 71, in 9 minutes 35 seconds, or at the rate of 7.4 fathoms per minute.

MANEUVERING TRIALS, SMITH.

Turning Trials.—The estimated diameter of the turning circle with helm hard-a-starboard was 350 yards, the time 2 minutes 44 seconds, the permanent angle of heel was 2 degrees to port. With helm hard-a-port the estimated diameter of the turning circle was 400 yards, the time about 2 minutes 54 seconds, permanent angle of heel from 1½ to 2 degrees to starboard.

The following times were taken at the steering wheel on the bridge:

Amidships to hard-a-starboard.....6.4 seconds
Hard-a-starboard to hard-a-port.....9.0 seconds
Hard-a-starboard to amidships.....4.4 seconds

The following times were observed at the tiller:

Amidships to 30 degrees starboard.....6.0 seconds
36 degrees starboard to amidships.....5.0 seconds
36 degrees starboard to 30 degrees port.....9.0 seconds
33 degrees port to amidships.....5.0 seconds

Backing Trials.—The helm being amidships and the vessel making approximately 28 knots, the engine telegraphs were thrown "full speed astern." The following times were taken, commencing when the signal to reverse was given:

Engines reversed..... 40 seconds
backing full speed.....1 minute 12 seconds
Ship dead in water.....1 minute 16.2 seconds

After the engines had been backing full speed about three minutes the signal for "full speed ahead" was given. The following times were taken:

Engines going ahead.....31 seconds
full speed ahead.....58½ seconds

REMARKS.

There is one particularly noticeable feature connected with these turbine installations, viz: the crowded condition of the auxiliaries in the engine room. As regards floor space, there is no noticeable improvement over the installation of reciprocating machinery.

In view of the necessity for keeping the auxiliaries in efficient condition and making them accessible for repairs and overhauling, it is suggested that there might be a pump room between engine and firerooms, with free communication to the engine room. Such an installation might require lengthening the vessel to some extent, but the resulting advantages would more than compensate. It could reasonably be expected that both the endurance and readiness for action of the machinery plant of any destroyer so fitted would be much greater than where the machinery is crowded together as at present.

The author's acknowledgments are due to Mr. J. W. DAWSON, of the office staff of the Inspector of Machinery, Cramp's shipyard, for the photographs illustrating this article.

INTERNAL-COMBUSTION ENGINES FOR NAVY LAUNCHES.

By Lieutenant W. G. Diman, U. S. N., Member.

Internal-combustion engines have been used for some time for driving launches, but only during recent years have they come into general use for small seagoing vessels. High speed and shallow-draught conditions both require the weight of ordinary propelling machinery to be greatly reduced in proportion to the power. In boats such as launches for ships' use, which must lie constantly waiting for a call to instant service, there is a marked advantage in the use of internal-combustion engines. The saving in weight and space is also such as to add greatly to the roominess and carrying capacity.

TESTS OF ENGINES.

In order to settle upon suitable types of internal-combustion engines for use in the naval service, a series of tests has just been completed at the U. S. Navy Yard, Norfolk, Va., on eighteen gasoline and oil engines of different types. The engines tested were fourteen four-cycle gasoline engines; one two-cycle two-port convertible kerosene and gasoline engine; one two-cycle three-port gasoline engine, and two two-cycle three-port oil engines. These engines as furnished by the manufacturers were first given a brake test in the shop, then placed in a forty-foot boat and given a thorough service test. A sample of the gasoline or oil fuel used on each test was analyzed, the gasoline on all tests running from 75 to 69.5 Baume. The engines were weighed before being tested, and the *weight per B.H.P.* calculated from this and the maximum B.H.P., obtained in the shop test. The plan at first was to

take the weight per rated B.H.P., but the ratings of the engines were found to be so indefinite that this was impraticable. The equipment placed in the boat was also weighed; this was practically the same for the gasoline equipments, but varied with the oil-engine equipment.

The gas and oil engines were tested as follows: The engine was placed on the testing slab, blocked up and connected by an intermediate shaft to a standard water brake. The engines were first run above rating and the revolutions then reduced by means of the brake. By running above, the greatest power that the engines could develop was obtained. There was found to be a characteristic B.H.P. curve with each of these gasoline engines, the horsepower increasing up to a certain point with the revolutions. Above that when the revolutions are increased the power will drop.

Measuring Fuel Consumption.—The fuel was placed in a tank having a capacity of 96 gallons and weighed accurately. The weight was then reduced to pints in order to obtain the *pints per B.H.P.* A pipe was led from the bottom of this tank to a separator, thence to the carburetor or oil pumps on the engine. This tank was filled with fuel and weight taken at the beginning and end of the test and the amount of fuel used per B.H.P. per hour calculated. Readings were also taken every ten minutes as a check on the flow of the fuel. With the gasoline equipment a thermometer was placed in the inlet manifold, and from this the temperature of the mixture entering the engine was obtained. This temperature was very important and it was found that 61 degrees F. was the most efficient temperature. The average fuel used per rated B.H.P. of all the engines was 1.199 pints per hour and at the maximum power was 1.216 pints.

Measuring Circulating Water.—The water used in the jackets was measured. Two large tanks were placed near the engines, each having a capacity of 305 gallons. Gauge glasses, properly marked with a scale to show the amount of water in each tank enabled the amount used to be read. The tanks were connected by a pipe with connection to suction for

circulating water in the jackets of the engine. On each tank was a plug cock so arranged that when one tank was empty it could be shut off and filled up. The temperature of the injection was taken at the tanks and the discharge temperature was taken just after it left the jacket. The exhaust led outside of the test room direct into the atmosphere, no muffler being attached. The temperature of the exhaust was taken by means of a pyrometer placed in an elbow outside of exhaust manifold. The true temperature of the exhaust was affected by the manifold being water jacketed and also by the water which was discharged into exhaust pipe. The average water used per B.H.P. on all the engines at the rated B.H.P. was 14.3 gallons per hour. There was no limit on the amount of water used, and a good average would appear to run about 11 gallons per B.H.P. per hour.

Measuring Lubricating Oil.—The lubricating oil was measured by means of a 5-gallon can which had been properly calibrated. When a test run started the position of the oil was noted in the oilers on the engine, and after a run, oil was taken out of the 5-gallon can to make up the deficiency and the amount noted. The lubricating oil, in pints per B.H.P. per hour, was calculated from this. The average amount of oil per rated B.H.P. per hour of all the engines was .047 pint. The engines were all new and used a greater quantity of lubricant than would be required in regular service.

Shop Tests, Gasoline Engines.—All thermomters and scales were calibrated. Each test run was of forty minutes duration in order to get the average conditions for the hour. During this time readings were taken every ten minutes. All results were figured for the hour.

The tests on gasoline engines were made as follows: First with the needle valve on carburetor set under normal conditions, the throttle wide open and the spark full advanced, the engine being run as high as the builders desired above normal rating. A run of forty minutes was made under these conditions, then the revolutions were reduced by means of the brake and another run of forty minutes made under the same conditions.

The revolutions were reduced again and another run made, and so on down until sufficient points had been obtained for a curve of B.H.P. and R.P.M. After these runs were made the throttle was moved in to $\frac{3}{4}$ and the spark advanced to $\frac{3}{4}$ and runs of forty minutes made under these conditions, starting from high revolutions and working down. This gave a curve for B.H.P. and R.P.M. under these conditions. The throttle was then moved in to $\frac{1}{2}$ and the spark advanced to $\frac{1}{2}$ and runs of forty minutes made under these conditions to obtain a curve for B.H.P. and R.P.M. The throttle was then moved in to $\frac{1}{4}$ and the spark advanced to $\frac{1}{4}$ and runs made under these conditions. After these runs were made the mixture was weakened by having the needle valve on the carburetor closed below normal conditions. The highest B.H.P. was taken from each of the above runs with a normal opening of the carburetor under the different conditions of spark and throttle and runs made with this mixture at the revolutions corresponding to the highest B.H.P. This gave a corresponding B.H.P. for each of the positions of spark and throttle and a comparision could then be made with the engine when running under normal conditions.

After these runs were made the mixture was strengthened by having the needle valve on the carburetor opened above normal conditions and runs made under the same conditions as on the weak mixture. This again gave a comparative result of the engine when running under normal conditions.

Tests Afloat.—On all the above runs the fuel and lubricating oil was measured. The engine was then installed in the boat. Runs were made on the 0.7-knot course off the Navy Yard for a speed curve at various revolutions. After this the engine was run in the boat for an hour under each of the following conditions; throttle $\frac{3}{4}$ open, spark $\frac{3}{4}$ advanced; throttle $\frac{1}{2}$ open, spark $\frac{1}{2}$ advanced; throttle $\frac{1}{4}$ open, spark $\frac{1}{4}$ advanced. This gave results comparative with the tests in the shop. The engine was then run in the boat for six hours steady, on full spark and throttle, under normal conditions of the carburetor. All the above runs were made in the boat on a displacement

of 13,200 pounds. If there was any material difference in the weight of the engine and 3,600 pounds (the weight allowed), this difference was removed from the boat and the boat run under this condition for a speed curve.

When these tests were concluded the engine was removed from the boat and opened up in order to measure it and also to note its condition.

Tests of Oil Engines.—With the oil engines the stroke of the pump was changed and runs made under these different conditions to get a B.H.P. and R.P.M. curve. With the convertible engine runs were made under practically the same conditions as with the gasoline engines except that no runs were made with the mixture changed. This engine was run on kerosene, except on the last runs which were made on gasoline in order to get a curve for a comparative result.

Results.—The power of the engines as shown in the brake tests varied from 30 to 74 H.P with revolutions at the rated H.P. varying from 350 to 1,000. The weight of the engines alone varied from 19.02 pounds per maximum B.H.P. to 81.8 pounds per maximum B.H.P. The average weight per maximum B.H.P. of the engines tested was 53.2 pounds.

From results obtained the average mean effective pressure of the engines tested running at about 500 R.P.M. was 77.7 pounds and the compression ran about 53 pounds. This was obtained from Mr. Frederick Groove's formula $M.E.P. = 2C - .01C^2$, in which $C =$ the compression pressure in pounds per square inch above the atmosphere. It was shown by the tests that the compression has a very important effect on the power developed by the engine for a given amount of fuel. When the motor is running on gasoline the absolute pressure in the cylinder after explosion is roughly $3\frac{1}{2}$ times the pressure in the mixture before explosion. A rise in temperature follows a rise in compression, and as the temperature increases the pressure increases. After the engine has been running for a certain definite time on a steady load the temperature becomes constant, assuming that the circulating water is contstant. Compression cannot be carried beyond a cer-

tain point for the heat developed would produce pre-ignition. Ninety pounds gauge pressure may be taken as a safe limit of compression for gasoline. High compression increases the power but also increases leakage. The cooling problem enters in, there is an increase in the load of the bearings, and the motor in general is apt to have a shorter life and be less flexible in its speed control.

CONCLUSIONS.

From the trials the following conclusions were deduced:

Most Suitable Type.—No gasoline engine with revolutions above 500 per minute should be installed in any heavy-duty boat. It might be better to limit the piston speed to 600 feet per minute. In all cases reliability and durability are of the first importance. In designing gasoline engines it is first necessary to know the conditions under which they are to operate. A small engine should be designed with a view to the greatest possible simplicity and a large engine with a view to economy. It is generally preferable to have the engine vertical, as the wear on the piston is more evenly distributed. A single-acting engine is the best construction, the simplest. The moving parts should be the same as in a standard marine steam engine. To box them up in the crank case in a heavy-duty engine is absurd. If a reliable engine is desired it must be a slow-running one that will have heavy and large parts to meet all requirements. Race-boat weights may run from 10 to 19 pounds per H.P., but this must not be considered a satisfactory machine for service conditions. Light high-speed engines for heavy and medium-duty boats are not satisfactory for continuous use; they will always give trouble and should never be placed in anything but a speed boat. For race boats running in inland waters where skilled men operate them, these light high-speed engines might be satisfactory up to a certain point, but when a boat and engine is needed to meet all conditions the slow heavy-duty engine is the only one that should be placed in a boat. The present tendency

is to install too light an engine. This is not good engineering. Speed and lightness must be sacrificed for reliability. For work boats the revolutions should run from 250 to 500 R.P.M., for medium-speed boats from 500 to 700 R.P.M., and for high-speed boats from 700 to 1,000 R.P.M. For all around work no four-cycle engine weighing less then 35 pounds per B.H.P. should be placed in any boat.

Framing, Casings. — All four-cycle heavy-duty engines should be of the open type. Accessibility of moving parts having a very high number of revolutions is not so necessary, as damage cannot be observed and adjustments cannot be made. It is dangerous for the attendant and should be avoided. It is correct under these conditions to make accessibility unnecessary by perfecting all parts, especially the lubricator. With slow-running engines accessibility to the crank case for examination and repair is the principal requirement. Openings in the side of the crank case are often obstructed by valve gear, carburetors, magneto, starting devices, etc. To examine the bearing parts they have to be, in a good many cases, taken down, a job which in many boats is tedious, and for large boats, under some circumstances, would mean practically a removal of the engine. The bed plate should be of cast iron and should act as a foundation for the engine. It should have webbed boxes cast in it for the crank-shaft bearings. It should be extended to carry the reverse clutch, and no clutch should be installed that is separate from the engine equipment. Columns or their equivalent should run from the bed plate to the cylinders, making the crank case as open as is possible.

Unnecessary Fittings.—There is a tendency on the part of all gasoline and oil-engine manufacturers to install a number of fittings which are not necessary for perfect operation. This increases complication without giving any better results in the engine. There should be nothing in connection with a gasoline-engine equipment that is not absolutely necessary to its running efficiency. The simpler the installation the

better. Less trouble will be experienced and less care will be necessary.

Cylinders.—The cylinders, if possible, should be cast singly, if not, they should be in pairs having a cylinder head to a pair of cylinders. If the cylinders are cast separately the circulating water should not pass through the cylinder gaskets but be by-passed from the cylinder barrels to the heads. The cylinders should be made of cast iron with plenty of jacket space.

Stroke and Diameter.—For ordinary running a stroke greater than the bore is preferred except in a high-speed engine for racing purposes. High-speed engines are generally built with a stroke equal to or less than the diameter of the bore, which will give better results than in slow-speed engines. Taking into consideration the propeller efficiency, the slow-speed engine with stroke larger than the bore is preferred.

Valves and Valve Gear.—The valves should be either automatically or mechanically operated. The inlet valves on four-cycle engines are of two types, the automatic and the mechanically-operated type. The exhaust valves are all of the mechanically-operated type. The automatic inlet valves are simple poppet valves opening into the cylinders. These valves work against a spring and have a small piston which acts as a guide working in a cap which is screwed into the top of the cylinder. These automatic valves do away with cams, push rods, etc., and at slow speeds prove very satisfactory. The spring in the valve is set for one speed, and any increase or decrease from the normal speed will decrease the efficiency at which the valves will work. These valves should be well oiled and the caps should never be allowed to become tight in the cylinder head. Mechanically-operated valves are in a general way worked from a cam on the cam shaft. These cams and cam shafts should be made large and heavy with a large bearing surface, and all bearings should be placed as near as possible to the cams. If mechanically-operated inlet and exhaust valves are used it is preferable to have them on opposite sides. In this method two cam shafts are necessary. In a small proportion of

four-cycle motors an auxiliary exhaust port is provided in the wall of the cylinder, which is uncovered by the motion of the piston just before the completion of the down stroke. By the addition of this auxiliary port the back pressure is greatly reduced. The gears for the cam shaft should be placed on the outside of the crank case, if possible, and protected by a casing which can be easily taken off and the gears examined. The push rods run up through guides, which should be made long to give a good bearing surface, and the tops of these guides should be cupped in order to hold a wick to properly lubricate the rod. These push rods should be made square so that they cannot turn in the guide. The guides for the valve stems should be made long and be water cooled. The seats for the valves should be water cooled as thoroughly as possible in order to prevent warping. The valves should be made of nickel or cast steel and the stems of high-grade carbon steel. The stem should be welded to the valve, and no screwing or pinning of the stem to the valve is satisfactory. The springs for the valves should be held on to the stem by means of a cupped collar held to the stem by a key.

Two-Cycle Engines.—The two-cycle is mechanically simpler, having no valves except in the two-port type and in the scavenging type. These engines are, as a rule, more complex in the travel of the mixture. With high-speed engines the two-cycle is more uncertain because the time allowed for the various events is so small. The mixture is pre-compressed in the crank case. The method of admitting the mixture into the crank case divides the motor into two classes, either of the two or the three-port type. With the two-port type an automatic valve is fitted in the crank case, allowing the mixture to go into the case. On high-speed engines it is difficult for this valve to close quickly. This valve should be very large and should be adjustable from the outside. To do away with this valve a third port is made in the cylinder wall just below the exhaust port. When the piston is at the end of the top stroke the mixture enters the crank case. In order to get as much of the mixture as possible into the

crank case a combination of the two and three-port type is sometimes used. The inlet ports must be made large and allow easy passage of the gas, as the time of transference of the gas is extremely short. Back-firing is prevented by a screen in the by-pass which is kept cool by the mixture flowing by it.

Pumps.—All pumps connected with the engine should be of the plunger type, being made amply large, of composition, and arranged to run slowly. There should be three pumps—one circulating, one bilge and one air pump. They should be driven either by gears, by an eccentric, or by a crank from the cam or crank shaft, and no friction, belt or chain drive should ever be used. These pumps should be made to run at slow speed, and it is a safe rule to limit the speed of the pump to half that of the engine. No bilge water should be allowed to go through the jackets. A check valve should be placed on the suction side of both the bilge and the circulating pumps, and these pumps should also be fitted with air chambers. With kerosene engines there should be a kerosene pump for each cylinder, so that each cylinder can be a unit in itself and not rely on one pump to supply fuel for all cylinders. No gear pump should be used in connection with any gasoline equipment.

The two most important factors in the running of gasoline engines is the mixture going to the engine and the ignition. If these two items are properly regulated the engine will ordinarily run satisfactorily.

Air Supply.—To secure an explosion in an engine running with gasoline it is necessary to have a certain minimum proportion of air. There is a maximum limit to this proportion in order to obtain ignition and combustion, and this limit will be reduced when it is desired to obtain the maximum power.

Fuel.—The finer gasoline is divided at the nozzle of the carburetor the better it will vaporize and the heavier will be the oil that can be used. Gasoline is midway between benzine and kerosene, and is sold in quantities according to its specific gravity, which on the Baume scale, runs from 60 to 70. Gasoline is ordinarily used for internal-combustion engines

in this country, but it is gradually becoming lower in proof, and it is only a question of time when the gasoline motor will develop into an oil motor. This, of course, will depend on the proof, the cost and the demand for gasoline.

Leaks.—There is danger in the use of gasoline which should never be lost sight of in working with it. All leaks found should be immediately repaired. Leaks will drain into the bilge, and as gasoline is lighter than water it will float and gas will be given off, increasing when the liquid is agitated by the motion of the boats. Unless this space is kept thoroughly ventilated an explosion is liable to result.

Carburetors.—There are various forms of carburetors on the market, but the simplest are generally the best. Carburetors on the two-cycle three-port type must be considerably larger than necessary for the four-cycle type of the same size and speed, and the auxiliary air valve must be light to decrease the sluggishness due to inertia. As fitted most carburetors are placed on the inlet pipe and take air for the carburetor from an opening on the carburetor itself, having no control of the temperature. This equipment is efficient for summer work when the weather is warm, but to meet all conditions of service an air pipe should lead from the air opening on the carburetor to a casing fitted on the unjacketed port of the exhaust pipe. In cold weather frosting in the inlet pipe and carburetor is an important factor, as it will form ice on the inside as well as the outside and gradually reduce the power of the engine.

Heating the Air.—When the motor is running the vapor pipe will gradually become cold and freeze due to the process of vaporization going on in the pipe. By heating the air to carburetor the engine will be more certain of operation in cold weather, but no more heat should be supplied than is just sufficient to effect the proper vaporization. The mixture should go into the cylinders at about 60 degrees F. The casing on the exhaust pipe should be fitted with holes and a baffle placed inside the casing, so that the air will have to travel all around the exhaust pipe before entering the air

pipe. A shutter should be fitted in this air pipe, which may be either opened or closed, depending on the temperature of the air, so that the air going into the carburetor may be at all times at the proper temperature. The inlet pipe from the carburetor to the inlet manifold should be fitted with a jacket, so that the warm circulating discharge from the engine jacket may pass through this jacket and warm the ingoing gas. Hot air may be taken from the air casing around the exhaust pipe instead of the water. The pipes to this jacket should be fitted with cut-out valves, so that in case the temperature runs up the warming of this jacket may be cut out. The inlet manifold on any engine should be so designed that the mixture will have an equal travel to all cylinders, so that the quality of the mixture and amount of gas will be the same to each cylinder.

Ignition, Magneto.—Ignition is the most important factor in a gasoline-engine equipment. It is generally of three types, high-tension jump-spark, low-tension jump-spark, and the make-and-break. The first two involve more electrical complications, while the latter—the make-and-break—is much simpler. With the make-and-break, however, the mechanical details enter. The make-and-break system is preferred in slow-running heavy-duty engines which run below 500 revolutions, while above that the low-tension jump-spark is preferred. The make-and-break contact points are now generally made of steel alloys. Blunt ends or flat surfaces give good contacts.

There is a system now in use which has the spark coil in connection with the plug. This is very simple and does away with much excess wiring, and also eliminates the high-tension current going to the plug. It is preferable to have two systems of ignition, using one as a reserve.

There is generally too little attention paid to ignition in order to get a successful performance of the machinery. The magneto is generally placed low down near the bilge of the boat. This should in all cases be raised up to a height with the tops of the cylinders to keep it out of dirt from the flooring. It should be placed on a good, stiff bracket and should

have a direct drive, no friction or belt drive being permitted. The magneto itself should be covered by some suitable material to keep it free from dirt and moisture.

The wiring for any high-tension work should be of heavy insulation, being at least 3/16 of an inch thick, and the shortest leads possible should be used. Wire for other work should be covered with good insulation having a para-rubber mixture. All wiring should be run in permanent conduits of either metal or rubber tubing, and these should be made perfectly watertight. The ends of all wires should be permanently marked and also the connections on the engine, magneto, switch, coil and timer (if used) to correspond, so that the ignition can be easily connected up. This is especially important in high-tension work. If a timer is used in connection with the electric ignition, it should be kept clean and the revolving contacts should firmly touch each contact point. All spark plugs should have a good, substantial barrel to screw into the cylinder with enough metal so that it will not spring.

Governor.—There should be a governor in connection with a gasoline engine which will control the opening of the throttle and the speed, so that when the clutch is thrown out the engine will not race. A satisfactory arrangement to prevent racing is to have a control arranged from the clutch handle so that when the clutch is thrown out it will automatically nearly close the throttle. The control handles should be worked on an arc and made long, giving a small travel of the control levers for a long travel of the handle.

Flywheel.—A large, heavy flywheel is an advantage in a slow-running heavy-duty engine, especially when the compression is high. This will aid in governing the engine and also give it a better balance. To relieve stresses in the crank shaft it is better to have the flywheel aft, but this will take up a great deal more room. For all around work it is better to place the flywheel forward. The hub on the flywheel should be split and held on the shaft by a key and bolts so that the flywheel will not become stuck fast to the shaft.

Lubrication.—The lubrication of a gasoline and oil engine is generally of three kinds: gravity feed, splash and forced feed. For high-speed engines the splash system of lubrication is preferred, especially when the engine is enclosed. With this system there is generally a small oil pump driven by the engine, drawing oil from the crank base and distributing it by means of piping to the various bearings and dropping again to the crank case. For slow heavy-duty engines the mechanical feed is preferred with as few feeds as possible. In this system a definite amount of oil under the direct control of the operator is delivered to each bearing. A simple method of oiling the cylinders is to run the gasoline and lubricating oil in the proportion of about one pint of oil to five gallons of gasoline. Two-cycle engines sometimes use grease cups on the crank-shaft bearings to prevent leakage. Another method is to place a large sight-feed oiler on the intake pipe. No chain, belt or friction drive should be used in connection with any mechanical oiler, but all drives should be direct.

Reversing.—There are three methods of reversing employed with a gasoline equipment in a boat; first, by reversing the engine alone; second, by reversing with a clutch; and third, by reversing with a reversible propeller. With the first method the engines are either of the two or four-cycle type with from four to six cylinders. Heavy-duty engines of this type of over 100 B.H.P. have been in successful operation for a number of years. Air is generally used in reversing. The reversing clutches (second method) are of three kinds, namely: the cone clutch, having two cone surfaces fitting into each other, the angle not being greater than 12 degrees to 20 degrees; the plate clutch, transmitting the friction through two or three plates and gradually taking up the load without a sudden shock; and the external expanding clutch, which is a ring of metal expanded by wedges against an internal surface of another thus making the friction at a greater distance from the center of the shaft. In a good many cases of late a positive lock is used in connection with the clutch

when running in the ahead motion. The load is taken up first by the friction clutch and then the positive lock is thrown in. When the horsepower is above 100 it begins to be difficult to move the reversing levers by hand. In using a reverse gear never overload the gear; it is preferable to have the gear too large than too small; the gear should always be kept properly adjusted and well lubricated. Care should be taken in throwing the reverse clutch in, and the load should be taken up as gradually as is possible in order to relieve the strain on the engine. In the third method the pitch of the blades is changed, the engine at all times running in the same direction. With small propellers the hub is naturally small and it is difficult to fasten the blades to the hub. This is the principal requirement, and if properly fitted it will wear in time. This device is light in weight and the operation of reversing is simple.

Types of Oil Motors.—Oil Motors are of two types, the first being where the oil is vaporized outside the cylinders by heat from the exhaust. In the first method it is necessary to start on gasoline and run until the exhaust pipe is hot, then the gasoline is cut off. The mixture, instead of passing direct from the carburetor to the cylinder, passes into a chamber surrounding the exhaust pipe, being so proportioned to properly vaporize the oil. This type can run on either gasoline or oil. The second method is where the oil is injected direct into the cylinders and vaporized then by the heat inside. The vaporizer in the cylinder is kept at a dull-red heat and liquid fuel is injected into the vapor. Before starting the vaporizer is heated by an external flame and it is afterwards kept hot by the explosion and compressions in the cylinders. In the Diesel type the compression runs up to about 500 pounds, and at the end of this stroke oil is forced into the compression space which is ignited and the burning is kept up gradually as the piston recedes. The fuel pumps are so timed as to inject oil into the cylinder when the piston is at the end of the compression stroke. The speed and power of the engine is regulated by the stroke of the fuel pumps.

LLOYD'S RULES FOR INTERNAL-COMBUSTION ENGINES.

Lloyd's Register of British and Foreign Shipping issued on June 16th, the following rules for the survey of internal-combustion engines for marine purposes:

GENERAL.

Section 1.—In vessels propelled by internal-combustion engines the rules as regards machinery will be the same as those relating to steam engines so far as regards the testing of material used in their construction and the fitting of sea connections, discharge pipes, shafting, stern tubes and propellers.

CONSTRUCTION.

Section 2.—The following points should be observed in connection with the design of the engines:

2. The shaft bearings, connecting-rod brasses, the valve gear, the inlet and exhaust valves must be easily accessible.

3. The reversing gear and clutch must be strongly constructed, and easily accessible for examination and adjustment.

4. In engines of above 60 brake horsepower which are not reversible, and which are maneuvered by a clutch, a governor or other arrangement must be fitted to prevent racing of the engine when declutched.

5. Efficient positive means of lubrication—preferably sight feed—must be fitted to each part requiring continuous lubrication.

6. The engines are of closed-in type; they must be so fitted that the contained lubricating oil can be drained when necessary, and in wood vessels an easily-drained metal or metal-lined tray must be fitted to prevent leakage of either fuel oil or of lubricating oil from saturating the woodwork.

7. Carburetors, where petrol is used, and vaporizers, where paraffin is used, should be so designed that when the engine is stopped the fuel supply is automatically shut off. If an overflow is provided in the carburetor or vaporizer, a gauze-covered tray with means of draining it must be fitted to prevent the fuel from flowing into the bilges.

Strong metallic-gauze diaphragms should be fitted, either between the carburetor (or vaporizer) and cylinders or at the air inlets.

8. If the ignition is electric, either by magneto or by coil and accumulator, all electric leads must be well insulated and suitably protected from mechanical injury. The leads should be kept remote from petrol pipes, and should not be placed where they may be brought into contact with oil.

The commutator must be enclosed, and the sparking coils must not be placed where they can be exposed to explosive vapors.

9. No exposed spark gap should be fitted.

10. In paraffin and heavy-oil engines, where lamps are used for ignition or for vaporizing, these lamps should be fixed by some suitable bracket, and the flame enclosed when in use.

11. The circulating-pump sea suction is to have a cock or valve on the vessel's skin placed on the turn of the bilge in an easily accessible position, and the circulating pipe is to be provided with an efficient strainer inside the vessel. The discharge overboard is to be fitted with a cock or valve on the vessel's skin if it is situated under or near the load line of the vessel.

12. A bilge pump worked by the engines, or an independent power-driven bilge pump is to be fitted to draw from each part of the vessel. In open launches this bilge pump may be omitted, provided suitable hand pumps are fitted.

13. The cylinders are to be tested by hydraulic pressure to twice the working pressure to which they will be subjected, the water jackets of the cylinders to 50 pounds per square inch, and the exhaust pipes and silencer to 100 pounds per square inch.

14. The exhaust pipes and silencer should be efficiently water-cooled or lagged to prevent damage by heat, and if the exhaust is led overboard near the water line, means must be arranged to prevent water from being syphoned back to the engine.

15. The machinery must be tried under full working conditions, the report stating the approximate speed of vessel, the number of revolutions of the engines at full power, both ahead and astern, and the lowest number of revolutions of the engines which can be maintained for maneuvering purposes.

RULES FOR DETERMINING SIZES OF SHAFTS.

Section 3.—The crank, intermediate and other shafts, if of ordinarily mild steel, are to be of not less diameters than as given in the following table. When special steel is used the sizes are to be submitted for consideration.

1. For petrol or paraffin engines for smooth water services:

Diameter of crank shaft, in inches, $= C\sqrt[3]{D^2S}$,
where D = diameter of cylinder, in inches,
S = stroke of piston, in inches.

Four-stroke cycle.	Two-stroke cycle.	Bearing between each crank.	Two cranks between the bearings.
For 1, 2, 3 or 4 cylinders.....	1 or 2 cyls.	$C = .34$	$C = .38$
6 cylinders.....	3 cyls.	$C = .36$	$C = .40$
8 cylinders.....	4 cyls.	$C = .38$	$C = .425$
12 cylinders.....	6 cyls.	$C = .40$	$C = .49$

For open-sea service add .02 to C.

Diameter of intermediate and screw shafts, in inches, $= C\sqrt[3]{D^2S(n+3)}$,
where D = diameter of cylinder, in inches,
S = stroke of piston, in inches,
n = number of cylinders.

For smooth-water services—	For open-sea services—
$C = .155$ for intermediate shafts.	$C = .165$
$C = .170$ for screw shafts fitted with continuous liners.	$C = .180$
$C = .180$ for screw shafts fitted with separate liners or with no liners.	$C = .190$

In engines of two-stroke cycle n is to be taken as twice the number of cylinders.

2. When ordinary deep-thrust collars are used, the diameter of the shaft between the collars is to be at least 21/20ths of that of the intermediate shaft.

3. In the cases of Diesel and other engines in which very high initial pressures are employed, particulars should be submitted for special consideration.

FUEL TANKS AND CONNECTIONS.

Section 4.—1. Separate fuel tanks are to be tested with all fittings to a head of at least 15 feet of water. If pressure feed tanks are employed, they are to be tested to twice the working pressure which will come on them, but at least to a head of 15 feet of water. If the tanks are made of iron or steel they should be galvanized.

2. Strong and readily removable metallic-gauze diaphragms should be fitted at all openings on petrol tanks.

3. Paraffin or heavy-oil tanks, not used under pressure, are to be fitted with air pipes leading above deck. Pressure-feed tanks and tanks containing petrol should be provided with escape valves discharging into pipes leading to the atmosphere above deck. The upper ends of all pipes are to be turned down, and pipes above 1-inch diameter are to be provided with gauze diaphragms at the end.

4. No glass gauges are to be fitted to fuel tanks containing either petrol, paraffin or heavy oil.

5. Filling pipes are to be carried through the deck so that the gas displaced from the tanks has free escape to the atmosphere.

6. Separate fuel tanks should be provided with metal-lined trays to prevent any possible leakage from them flowing into the bilges or saturating woodwork. Arrangements are to be provided for emptying the tanks and draining the trays beneath them. For petrol tanks the trays must have drains leading overboard where possible, or they should be gauze-covered trays with means for draining them.

7. All fuel pipes are to be of annealed seamless copper, with flexible bends. Their joints are to be conical, metal to metal. A cock or valve is to be fitted at each end of the pipe conveying the fuel from the tank to the carburetor or vaporizer. The fuel pipes should be led in position where they are protected from mechanical injury and can be exposed to view throughout their whole length.

8. The engine room and the compartment in which the fuel tanks are situated are to be efficiently ventilated.

9. An approved fire-extinguishing apparatus must be supplied.

PERIODICAL SURVEYS.

Section 5.—1. The machinery is to be submitted to survey annually. At these surveys the cylinders, pistons, connecting-rods, crank and other shafts, inlet and exhaust valves and gear, clutches, reversing gear, propeller, sea connections and pumps are to be examined. The electric ignition is to be examined and the electric leads tested. The fuel tanks and all connections are to be examined, and, if deemed necessary by the surveyor, to be tested to the same pressure as required when new. If practicable, the engines should be tested under working conditions.

2. The screw shaft is to be drawn at intervals of not more than two years.

U. S. S. *PRESTON.*

DESCRIPTION AND OFFICIAL TRIALS.

By William Ashby Leavitt, Jr., Associate.

The *Preston* (Torpedo-Boat Destroyer No. 19), built by the New York Shipbuilding Company, Camden, New Jersey, is one of the five vessels of this class authorized by Act of Congress making appropriations for the fiscal years ending June 30th, 1907, and June 30th, 1908. The remaining vessels of this class have been already described in the JOURNAL; the *Flusser* and *Reid* in the February, 1910, number, and the *Smith* and *Lamson* in the present issue.

The contract was signed September 28th, 1907, time for completion to be twenty-four months; the keel was laid April 29th, 1908; vessel launched July 14th, 1909, and delivered to the Government at the Philadelphia Navy Yard December 21st, 1909. The delay of two months and twenty-three days over the contract time was occasioned by difficulty in securing sound castings for the turbine engines, much of the work being condemned after considerable progress had been made in the machine shop.

The contract price for the construction of the hull and machinery was $645,000.00, of which $270,000.00 was allotted to the hull and $375,000.00 for propelling machinery and auxiliaries.

The contract required a progressive trial over a measured mile in water not less than 30 fathoms deep, for the purpose of standardizing the screws, the runs extending from maximum to a minimum speed of 12 knots, at a displacement of 700 tons; a full-speed trial of four hours' duration in open sea in deep water, at a displacement of 700 tons, at an average speed of not less than 28 knots per hour, the steam pres-

sure not to exceed 240 pounds per square inch, and the water consumption not to exceed 15.5 pounds; an endurance and coal-consumption trial of twelve hours in open sea in deep water at a displacement of 700 tons, at an average speed of not less than 24 knots per hour, the speed to be as nearly uniform as possible, and the water consumption not to exceed 17 pounds; an endurance and coal-consumption trial of twenty-four hours in open sea in deep water (which trial was reduced to twelve hours by direction of the Navy Department), at a displacement of 700 tons, at an average speed of 16 knots, the speed to be as uniform as possible, and the average water consumption not to exceed 26.75 pounds. All of the above-mentioned trials were run off successfully, and the contractors are deserving of full credit for having turned out such an excellent piece of work, especially in view of the fact that the vessel was the first destroyer built by them, and contained their first attempt at the manufacture of Parsons turbine work, embodying such intricate and delicate construction, all of which worked most satisfactorily under trial.

GENERAL DESCRIPTION OF THE VESSEL.

Hull.

The hull is built of medium and hard steel, galvanized in all parts of shell and frame below the water line, and all parts lightened by omitting or by cutting out useless material. The proportion of length to beam of hull is 11.14 to 1.

There is a main deck and raised forecastle, the latter extending as far aft as frame 41. Under the forecastle are located the quarters for four commissioned officers, a bath room and toilet, the officers' pantry, the ship's galley, and a crew's water closet.

There are four smoke pipes, oval in section, raised hatch coamings for the firerooms, four forced-draft down-takes with concave hoods which can be raised to form deflectors for the wind or can be battened down to keep out the sea or weather. The air ducts are about waist high when closed.

Abaft the smoke pipes on deck is the engine-room hatch, abaft of which is the deck house, including a crew's water closet and wash room, a wireless-telegraph room and a hatchway to the berth deck.

Under the forecastle forward of the officers' quarters, from frame No. 12, is the lamp room, and forward of that is a small storeroom in the bow. The berth deck extends as far aft as frame 48 and aft from frame 124 to frame 160. Between frames 16 and 48 are the petty officers' quarters and the crew space, separated by a watertight bulkhead at frame 29, with a watertight door, the petty officers' space being forward of the crew's space. The latter is reached from the main deck by a ladder, and the petty officers' space is reached from forecastle deck also through the crew space. Forward of the crew space on the berth deck are an electrical and equipment storeroom and the paint and oil storeroom. A smaller storage space is available in the bow. The narrow space in the bow is filled in with concrete of coke and cement. The two forward storerooms are reached from the deck above, and the electrical storeroom is reached through a door in the forward watertight bulkhead of the petty officers' space. These living spaces are sheathed with No. 27 U.S.G. sheet iron, cork painted. In the petty officers' quarters are fourteen metal folding berths, and in the crew space are thirty-six of these berths. One mess table is provided for the petty officers and two for the crew space. As the crew space is adjacent to the forward boiler space, the bulkhead between them is insulated by one inch of asbestos fire-felt, sheathed by No. 27 sheet iron, on the after side. These quarters and storerooms are lighted by dead lights and air ports in the ship's sides. There is no artificial ventilation excepting electric fans in living spaces.

The after berth deck is divided as follows: From frame 124 to 138 are the quarters for 27 men of the engineer's force. From 138 to 149 are quarters for 8 chief petty officers. Metal folding berths for the engine-room force are arranged three tiers high, and for the chief petty officers two tiers high. There

is one mess table in the latter and two in the former compartments. Abaft of the chief petty officers to the stern is a construction and repair storeroom. Under the engineer-crew space are three ammunition rooms, a magazine for war heads, a handling room and the shaft alley. Under the chief petty officers' quarters are the engineers' and ordnance storerooms. Under the construction and repair storeroom are two trimming tanks of the following capacities: D-6, 8 tons salt water; D-7, 4.9 tons salt water. From frame 154 to the rudder port the stern framing is cut away to form a recess for the middle propeller. The living spaces and storerooms on the berth-deck aft, like those forward, are provided with air ports and dead lights.

Under the crew space forward, from frame 40 aft, are a fresh-water tank, a handling room and 3-inch magazine, reserve feed-water tank and wet-provision storeroom. Under the petty officers' space are the dry provisions, officers' mess stores and small storeroom. Next comes the chain locker (under the electrical storeroom), and next (under the paint and oil room) is a 9.8-ton trimming tank, forward of which, in the fore foot, is a 1.8-ton trimming tank.

There are two boiler compartments and one engine compartment.

The capstan is located on the forecastle with its engine secured immediately underneath on the watertight bulkhead between the captain's and the lamp rooms. It is a diminutive vertical-cylinder (two) steam engine, built by the Hyde Windlass Company, Bath, Maine, and is controlled by a lever from deck.

The machinery spaces extend from bulkhead at frame 49 to frame 124, exclusive of the shaft alley, which latter extends as far aft as frame 138. The engine room is included between frames 97 and 124.

Coal bunkers extend alongside the boiler spaces. In addition to these fore-and-aft bunkers there are two athwartship bunkers, one between the boiler compartments and one between the after boiler compartment and the engine room.

The following are the particulars of the hull:

Length between perpendiculars, feet and inches	289–00
Projection forward of F.P., feet and inches	2–02
aft of A.P., feet and inches	2–08½
Length over all, feet and inches	293–10½
on load water line, feet and inches	289–00
of straight keel, feet and inches	242–07⅜
Breadth, molded, at 8 ft. 0 ins. water line, feet and inches	25–11
Depth, molded, main-deck side at frame 83, feet and inches	16–04 3/16
Draught, normal, feet and inches	8–00½
Displacement, normal, tons	695.4
per inch at designed draught of 8 ft. 0½ in., tons	11.8
Ratio of length to beam	11.4
Area L.W.L., plane, square feet	4,971.0
of wetted surface, square feet	7,600.0
C.G. of L.W.L., plane, aft of M.S., feet	2.46
C.B., above bottom of keel, feet	5.16
C.B., aft of M.S., feet	0.122
Transverse metacenter above C.B., feet	8.03
Longitudinal metacenter above C.B., feet	817.88
Coefficient of fineness, block	0.403
midship section	0.652
L.W.L.	0.662
Number of frames, spaced 21 inches	164

COMPLEMENT.

Commanding officer,	1
Officers,	3
Seaman branch,	24
Artificer branch,	51
Special branch,	2
Commissary branch,	2
Messmen branch,	3
Total,	86

CAPACITY OF TANKS.

Compartment.	Gallons.	Tons F. W.	Tons S. W.
Gravity tank	50	...	...
Reserve-feed tank A–10	1,878	7.0	...
D–1	1,153	4.3	...
Trimming tank A–1	480	1.8	1.8
A–2	2,573	9.8	9.6
D–6	2,100	8.0	7.8
D–7	1,290	4.9	4.8
Fresh-water tank A–6	3,515	13.0	...

HEIGHTS.

Above outer keel plate amidships:

Top of truck foremast, feet and inches	67–10¼
mainmast, feet and inches	68–03
Searchlight platform floor, feet and inches	38–08 3/16
Bridge at side, feet and inches	31–00 1/16
Top of forecastle deck at stem, feet and inches	25–04½
main deck at stern	16–04 3/16

Admission and egress to and from the boiler spaces is by raised hatches, one over each side of fireroom. These hatches have spring catches and glass dead lights.

MAIN ENGINES.

The propelling machinery consists of Parsons turbines, a description of which, covering the general features, is given below:

The cylinders or casings are made of hard close-grained cast iron, divided into two parts at the axis on a horizontal plane, the bearings at each end being a part of the lower casting, this lower half also being provided with feet for bolting to the seatings. The method of machining the cylinders was to rough-bore and plane, after which the parts were bolted together and given the maximum hydrostatic test. Castings were then heated by steam at atmospheric pressure for 24 hours, to anneal same, and allowed to cool, after which the finishing work was done and the cylinders grooved for the blading.

The rotor drums are Class A forged steel, and the rotor wheels, or spiders, are forged in one piece, being a part of the rotor shaft; the shafting being Class A forged steel, and the wheel portion Class B forged steel. These forgings were received at the works rough machined, the spiders being slotted out to true proportions, thus giving a perfectly balanced rotor —a very expensive operation, but undoubtedly the most satisfactory when finished as in this instance. The blading is Parsons standard system, the composition of the blades and binding strips being 72 per cent. copper and 28 per cent. zinc;

the caulking strips being 63½ per cent. copper and 36½ per cent. zinc, and the binding wire of copper.

The arrangement provides for five ahead and two astern turbines (the astern turbines being a part of the L.P. ahead), driving three shafts for going ahead and two when going astern, the disposition of the turbines being as follows: The center shaft being driven by the main H.P. turbine, the starboard shaft by the I.P. cruising, and one L.P. ahead turbine and the port shaft by the H.P. cruising and one L.P. ahead turbine, the cruising turbines for both of the wing shafts being forward of the low-pressure turbines. For full speed ahead steam is admitted to the main H.P. turbine, and after expanding into the two L.P. turbines is exhausted into the condensers, the H.P. and I.P. cruising turbines under this condition revolving in a vacuum, this vacuum being maintained through the turbine drains.

For low cruising speeds steam is admitted to the H.P. cruising turbine and expanded into the I.P. cruising turbine, and from this turbine into the main H.P. turbine, and thence through the L.P. turbines into the condensers.

For high cruising speeds steam is admitted to the I.P. cruising turbines and expanded into the main H.P. turbine and thence through the two L.P. turbines into the condensers; steam not being admitted to the H.P. cruising turbine under this condition, the turbine revolving in a vacuum which is maintained through the drain, the connection between the H.P. cruising and L.P. cruising turbines being closed by means of a self-closing valve. Self-closing valves are also fitted on the exhaust pipe from the main H.P. to the L.P. turbines, so that the L.P. ahead and the astern turbines can be operated as desired, the starboard and port engines being entirely independent. Under these conditions the main H.P. turbine is not in use and is connected to the condenser by the drain. The exhaust pipe from the H.P. cruising to the I.P. cruising turbine and from the I.P. cruising to the main H.P. turbine is provided with screw-down self-closing valves.

MAIN-ENGINE DATA.

Diameter of rotor drums :

Main H.P. turbine, inches	29½
H.P. cruising turbine, inches	30
I.P. cruising turbine, inches	29
L.P. ahead turbines, inches	43½
L.P. astern turbines, inches	29½

Diameter of cylinders :

Main H.P. turbine, inches	32⅝	34	35½	38		
H.P. cruising turbine, inches	30⅝	31	31⅜			
I.P. cruising turbine, inches	30⅝	31½	32			
L.P. ahead turbines, inches	46½	48	49½	52	54	59
L.P. astern turbines, inches	31½	32½	35			

Number of expansions :

Main H.P. turbine	4
H.P. cruising turbine	3
I.P. cruising turbine	3
L.P. ahead turbines	8
L.P. astern turbines	4

Length of cylinder for blading, for each diameter and expansion noted above :

Main H.P. turbine, inches	13½	14⅝	17	18½			
H.P. cruising turbine, inches	16½	17½	17⅝				
I.P. cruising turbine, inches	16⅝	21⅝	20½				
L.P. ahead turbines, inches	7	7½	3 of 8	8½	8⅝	8½	
L.P. astern turbines, inches	9⅝	11½	14½	15			

Rows of blading for each expansion :

Main H.P. turbine	10
H.P. cruising turbine	18
I.P. cruising turbine	16
L.P. ahead turbine	4 of 5, 4 of 4
L.P. astern turbine	8

Length of blades for each expansion :

Main H.P. turbine, inches	1⅝	2½	3	4½		
H.P. cruising turbine, inches	0⅝	0½	0 1/16			
I.P. cruising turbine, inches	0⅞	1½	1½			
L.P. ahead turbine, inches	1⅝	2½	3	4½	5½	3 of 7½
L.P. astern turbine, inches	0⅝	1⅝	2⅝	2⅝		

Rotor shafting and bearings :	Length.	Diameter.	Diameter axial hole.	Length over all of rotor drum and shaft.	
	ins.	*ins.*	*ins.*	*ft.*	*ins.*
Main H.P. turbine	7	7	3½	11	11 1/16
H.P. cruising turbine	7	7	3½	10	02 3/16
I.P. cruising turbine	7	7	3½	10	10⅝
L.P. ahead turbine	12	7	3½	17	08 1/16
L.P. astern turbine	12	7	3½	...	

Thrust bearings:

Collars on shaft:	M.H.P.	H.P.C.	I.P.C.	L.P. (one).
Number	16	8	8	16
Thickness, inch	0½	0½	0½	0½
Space between, inch	0⅝	0⅝	0⅝	0⅝
Outside diameter, inches	8¾	8¼	8¼	8¾
Inside diameter, inches	5¼	5¼	5¼	5¼
Bearing surface, top, sq. inches.	265.2	101.12	101.12	265.2
bottom, sq. in..	282.88	115.56	115.56	282.88
Number of shoes, top	15	7	7	15
bottom	16	8	8	16

Effective thrust surface (oil grooves out), full speed ahead, sq. in..... 736.5
astern, square inches..... 480.0
cruising speed, sq. inches... 736.5

Steam thrust, square inches.....201.87 + 736.5 = total, 938.37

Number and diameter of steam and exhaust connections:

	M.H.P.	H.P.C.	I.P.C.	L.P. ahead.	L.P. astern.
Steam, inches.....	2 of 9	1 of 5	1 of 7½	1 of 6	1 of 7
Exhaust, inches..	2 of 16	1 of 9	1 of 13	1 of 39½ × 46⅝	

The working platform is immediately abaft the forward engine-room bulkhead, upon which is located all of the valves and other operating gear, such as gauges, counters, telegraphs, etc. Composing the group of turbine-operating valves are the 9-inch angle; main steam-stop valves, one on each side; the starboard valve admitting steam from the two forward boilers, and the port valve from the two after boilers. Pipes from these valves lead to a distributing casting located amidships, at the bottom of which is a 12-inch balanced throttle valve, at the bottom of which is a steam strainer with two 9-inch branches for steam to the main H.P. turbine, and on the starboard side of this valve is located a 7½-inch globe stop valve for steam to the I.P. cruising turbine, and on the port side a 5-inch globe stop valve for steam to the H.P. cruising turbine.

On the horizontal arms of the distributing casting are two combination double-beat valves for steam direct to the L.P. ahead and astern turbines, these valves being used for maneuvering; the valve stems being actuated by a double-ended lever controlled by springs, and so arranged that when one valve is opened the other is closed. The upper valve of the combination is 7-inch for steam to the L.P. astern turbines and the bottom valve is 6-inch for steam to L.P. ahead turbines. By

turning the maneuvering valve hand-wheel to the right the lower valve is closed and the upper one is opened, thereby admitting steam to the L.P. astern turbine, the operation, of course, being vice versa for going ahead.

All valves and castings are composition and all piping is of seamless-drawn steel, with rolled-steel flanges.

SHAFTING.

There are three lines of shafting, all of Class A forged steel, arranged as follows: the starboard and port lines aft of the turbines each consisting of four sections, two of which are line, one stern-tube and one propeller; and the center line consists of five sections, four of which are line and one stern tube, the latter in this instance also being the propeller shaft. The shaft couplings are of the Lovekin type described many times in this JOURNAL. The stern-tube shafting is the only portion that is covered, the casing at the bearings being of composition H, and that between the bearings being seamless-drawn brass tubing $\frac{1}{8}$-inch thick.

Data for Shafting.

Outboard shafting:	
Line shafting, forward section, length, feet and inches	12–01 $\frac{7}{16}$
after section, length, feet and inches	12–03
Stern-tube shafting, length, feet and inches	22–02 $\frac{1}{2}$
Propeller shafting, length, feet and inches	17–02 $\frac{11}{16}$
Total length, feet and inches	63–09 $\frac{3}{8}$
Center shafting:	
Line shafting, forward section, length, feet and inches	21–01 $\frac{3}{16}$
center, length, feet and inches	18–00
after center, length, feet and inches	16–06
section, length, feet and inches	14–00
Stern tube, length, feet and inches	19–00 $\frac{7}{16}$
Total length, feet and inches	88–07 $\frac{3}{8}$
Diameter of shafting, inches	00–06 $\frac{1}{2}$
axial hole, inches	00–03 $\frac{1}{2}$
coupling flanges, inches	00–11 $\frac{3}{8}$
Thickness of coupling flanges, inches	00–01 $\frac{3}{8}$
Diameter of shafting at steady bearings, inches	00–07 $\frac{1}{2}$
Length of steady bearing, inches	00–09 $\frac{1}{2}$

Number of steady bearings:	
Outboard shafting, each	2
Center shafting	5
Length of inboard stern-tube bearings, feet and inches	1–09
outboard stern-tube bearings, feet and inches	2–09
strut bearings, feet and inches	2–09

PROPELLERS.

The propellers are of manganese-bronze, the blades being cast solid with the hub, which is bored taper to fit the propeller shaft, being held in place by two longitudinal keys and a composition nut on the end of the shaft. The nut is locked in place and covered by a composition cap bolted watertight to the hub. The driving faces of the blades were machined to a true surface, the backing faces being ground to a good surface, after which the entire surface was polished. Each propeller was accurately balanced. The starboard and outer propeller are right-handed and the port left-handed.

Data for One Propeller (Regular).

Number of blades	3
Diameter of propeller, feet and inches	5–00
Pitch of propeller, feet and inches	4–09
Ratio of diameter to pitch	0.95
Area, projected, square feet	12.27
helicoidal, square feet	13.44
disc, square feet	19.635
Ratio of projected to disc area,	0.625
Area of immersed midship section, square feet	136.0
Ratio disc to immersed midship-section area	0.144
projected disc to immersed midship-section area	0.0902
Height of lower tip of blade above keel, wing, inches	0–03
below keel, center, inches	0–14½
Immersion of upper tip of blade, wing, feet and inches	2–10½
center, feet and inches	4–03
Length of longitudinal keys, inches	0–13½
Thickness of longitudinal keys, inch	0–00¾
Width of longitudinal keys, inches	0–01½

Spare propellers made subsequent to the trial have the following dimensions:

Diameter, feet and inches	5–03⅝
Pitch, feet and inches	4–10
Ratio of diameter to pitch	0.913
Area, projected, square feet	13.56
helicoidal, square feet	15.16
disc, square feet	21.58
Ratio of projected to disc area	0.628

BOILERS.

There are four boilers of the Thornycroft express type, coal burning, two in each boiler compartment, each boiler having a separate smoke pipe. Each boiler has a single furnace, with four furnace and two ash-pit doors, the firerooms being closed when under forced draft. The ash-pit doors are in-swinging and arranged to close automatically in case of a tube bursting. The furnace doors, although of the out-swinging type, can be closed easily by hand, as they are overbalanced.

The boilers consist of the usual steam and water drum located fore and aft, and almost entirely surrounded by the uptake, about 10 feet 9 inches above the base line of the boiler, below which, on either side, are the mud drums, 2 feet 7 inches above the base and spaced 6 feet 6½ inches from the center line of the boiler. The down-take tubes, one on each side, are at the forward end of the boiler, back of which are the generating tubes placed tangent to each other, but arching over to meet the steam drum in such a manner as to form a spacious combustion chamber and grate area.

The casings are of light galvanized-iron plate, stiffened in an approved manner, that for the sides consisting of two plates, the outer being No. 16 gauge lined with ¼-inch asbestos board, and the inner of No. 12 gauge, between which and the asbestos board is a 2-inch air space. The front and back casings consist of two plates, the outer and inner respectively being No. 7 and No. 16 gauge, a 6-inch air space being provided for the front and a 3-inch for the back, the outer plate being provided with air doors, in-swinging, so as to close easily if any undue pressure occurs from within. The inner

side of the inner plates is lined with fire brick 2½ inches thick, to a point about 2 feet 3 inches above the grate level, where it is five inches thick. Wrought-iron grate bars were used on the trial but have been replaced by standard cast-iron bars, since delivery to the Government.

Particulars and Dimensions of Boilers (each).

Number of boilers (two in each boiler compartment)	4
Designed working pressure, pounds	250
Hydrostatic test applied, pounds	375
Steam-pressure test applied, pounds	300
Grates, length, feet and inches	7–11½
width, feet	11–00
Per cent. of air space in grates	64.19
Grate area, each, square feet	87.5
Total grate area, square feet	350.0
Number of furnaces	1
Number of furnace doors	4
Heating surface (one boiler), area in square feet	4,800.0
Total heating surface, all boilers, in square feet	19,200.0
Ratio of grate to heating surface	1 to 54.9
External height of boilers, feet and inches	12–09¾
width of boilers, feet and inches	15–07
length of boilers, feet and inches	12–07
Tubes, outside diameter, inches	1⅛ and 1¾
thickness, B.W.G., No.	11 and 12
kind	Cold-drawn seamless steel.
number to each boiler (146 = 1⅜, 1,816 = 1¼)	1,962
longest, feet and inches	9–04½
shortest, feet and inches	7–11⅞
Smoke pipes, number	4
height above grates, feet and inches	*25–03½
area of section, each, square feet	11.0
Ratio of grate area to area of each smoke-pipe section	7.9 to 1
Drum, upper, diameter, inside, inches	42
thickness, top sheet, inch	$00\frac{37}{64}$
lower sheet, inches	$01\frac{9}{16}$
Upper drum heads are bumped to a radius of, inches	42
thickness, inch	00¾
Lower drums, diameter, inside, inches	19
thickness, inch	$00\frac{37}{64}$

* Eight feet has been added to the height of smoke pipes since delivery to the Government.

MAIN CONDENSERS.

There are two condensers, located one on each side of the engine room. They are cylindrical, with shells of wrought-steel, water chests of copper, tube sheets of naval brass, and tubes of composition 70 per cent. copper, 1 per cent. tin and 29 per cent. zinc.

The tubes are straight, fixed at one end by expanding into tube sheets, and at the other end are provided with screw glands and packing.

The circulating water is furnished by independent circulating pumps discharging directly into the short water chest or nozzle at forward end of condenser, thence passing through the tubes and overboard through a similar nozzle.

Data for One Condenser.

Length of shell over tube sheets, feet and inches	12–02
Inside diameter of shell, feet and inches	4–06½
Thickness of shell, inch	0–00½
tube sheets, inch	0–01
Number of tubes	2,067
Outside diameter of tubes, inch	0–00⅝
Thickness of tubes B.W.G., No	18
Length of tubes, feet and inches	12–02 1/16
Number of stay tubes	7
Pitch of stay tubes, inches	0–12½
Size of stay tubes, steel pipe, inches O.D	0–01⅝
gauge B.W.G	0–08
Cooling surface, measured on outside of tubes, square feet	4,051.0
Ratio of total cooling surface to total heating surface	0.422
Area through tubes, square inches	451
circulating-water inlet and outlet, square inches	254.47
Ratio tube area to circulating inlet and outlet	1.77

The following connections are on each condenser shell:

One main turbine exhaust, rectangular, square feet	13.0
auxiliary exhaust, inches	6
bleeder, inches	2½
feed-heater drain, inches	1½
air-pump suction, inches	6½
safety valve, inches	2
boiling-out connection, inch	0¾

Auxiliary.	Number.	Type and make.	Steam cylinders.				Water cylinders.			
			Number.	Diameter, inches.	Dia. piston rod, inches.	Stroke, inches.	Number.	Diameter, inches.	Dia. piston rod, inches.	Stroke, inches.
Main air pumps	2	Blake, vertical, simplex, beam	1	12	2¼	18	2	26	2	18
Main circulating pumps	2	N. Y. S. Co., vertical, single-engine centrifugal pump	1	8	1⅞	8	1	31	im-peller	18 inlet. 18 outl't.
Main feed pumps	2	Blake, vertical, simplex, piston	1	12	2¼	16	1	8½	2¼	16
Auxiliary feed pumps	2	Blake, vertical, simplex, piston	1	12	2¼	16	1	8½	2¼	16
Engine-room fire and bilge pumps	1	Blake, vertical, duplex, piston	2	7½	1½	8	2	5	1½	8
Fireroom f. and b. pumps	2	Blake, vertical, duplex, piston	2	7½	1½	8	2	5	1½	8
Aux. condenser pumps	1	Blake, horizontal, simplex, combined air and circulating	1	6	1¼	7	1 a 1 w	8 8	1¼	7
Distiller fresh-water pumps	1	Blake, vertical, simplex, piston	1	3½	0⅞	4	1	4	0⅞	4
Evaporator feed pumps	1	Blake, vertical, simplex, piston	1	3½	0⅞	4	1	4	0⅞	4
Oil-circulating pumps	2	Blake, vertical, simplex, piston	1	6	1 7/16	8	1	7	1 7/16	8
Oil-cooler circ'lat'g pumps	1	Blake, vertical, simplex, piston	1	6	1 7/16	8	1	5½	1 7/16	8
Forced-draft blower eng's	4	Forbes, horizontal, double-engine	2	4½	1 7/16	5	...	...	...	...
fans	4	Sirocco, 40-in. diameter, blades 17 in., air inlet 34⅛ in., max. revolutions, 920.								
Air compressors	2	Mahn type, steam, vertical, model 1. Bliss Mfg. Co., Brooklyn, N. Y. Capacity, 25 cubic feet.								
Dynamo turbines	1	Curtis, turbo generator, single-stage, direct connected, 5 kw., 5,000 r.p.m., 125 volts.								

PUMPS AND CONNECTIONS.

Pumps.	Ins.	Suction pipes from:	Ins.	Discharge pipes to:	Location.
Main air	10	Condenser	9	Feed tanks	After-end engine room.
circulating	18	Sea	18	Condenser	
	7	Bilge	...		Engine room.
feed	6	Feed suction, feed and filter tank and reserve-feed tanks.	5	Main feed discharge	Engine room.
	4½	Channel ways	...		
Auxiliary feed	4½	Feed suction and reserve-feed tanks.	2½	Auxiliary feed	One in each boiler room.
	2	Bottom blows	2	Overboard	One in each boiler room.
Engine-room fire and bilge.	3½	Sea	2½	Fire main	In engine room, aft.
	4	Bilge in engine room	3	Overboard	
	3	C. and R. drainage	1½	Distiller	
	1½	Hose connection	1½	Hose connection	
Fireroom fire and bilge	4	Fireroom bilge	2½	Fire main	One in each boiler room.
	3½	Sea	3	Overboard	One in each boiler room.
	3	C. and R. drainage	2	Ash ejector	
	1½	Hose connection	1½	Hose connection	
Auxiliary condenser	4	Sea	3	Main feed tank	In engine room, forward.
	4	Auxiliary condenser	4	Overboard	
Distiller fresh water	1½	Distiller	1	Reserve-feed tanks	In engine room, aft.
			0½	Ship's tanks and feed and filter tanks	
Evaporator feed	1½	Sea	1	Evaporators	In engine room, aft.
	1½	Fire and bilge discharge overboard	...		
Oil circulating	2½	Oil drain tank	2½	Through cooler to bearings.	In engine room, forward.
cooler circulating	3½	Sea	3	Through cooler, overboard.	In engine room, forward.

The following connections are on each water chest:

One main injection on forward chest, inches	18
outboard delivery on after chest, inches	18
manhole, inches	12
drain, inches	00½
air cock, inches	01
thermometer, inches	01

DATA FOR ONE CIRCULATING PUMP.

Capacity of pump, gallons per minute	10,000
Diameter of suction nozzle, inches	18
discharge nozzle, inches	18
impeller, inches	31
Width of tip of blade, inches	03½
Diameter of engine cylinder, inches	08
Stroke of engine cylinder, inches	08
Diameter of piston rod, inches	01¾

WEIGHTS OF PROPELLING MACHINERY.

The actual weight of the propelling machinery and auxiliaries is 255.114 tons, as follows:

	Pounds.
Main engines	98,693
Shafting	20,294
Main condensers	27,718
air and circulating pumps	20,116
Propellers	3,814
Boilers	136,714
Boiler fittings	40,420
Smoke pipes and uptakes	5,807
Steam and exhaust piping and valves	41,009
Suction and discharge piping and valves	24,556
Lagging and clothing	15,870
Flooring, gratings, handrails, etc	8,503
Auxiliary machinery	23,125
Fittings and gear	10,678
Tools and spare parts on ship	13,846
Miscellaneous machinery	19,296
Water	60,999
Total	571,458
Total weight in tons	255.114

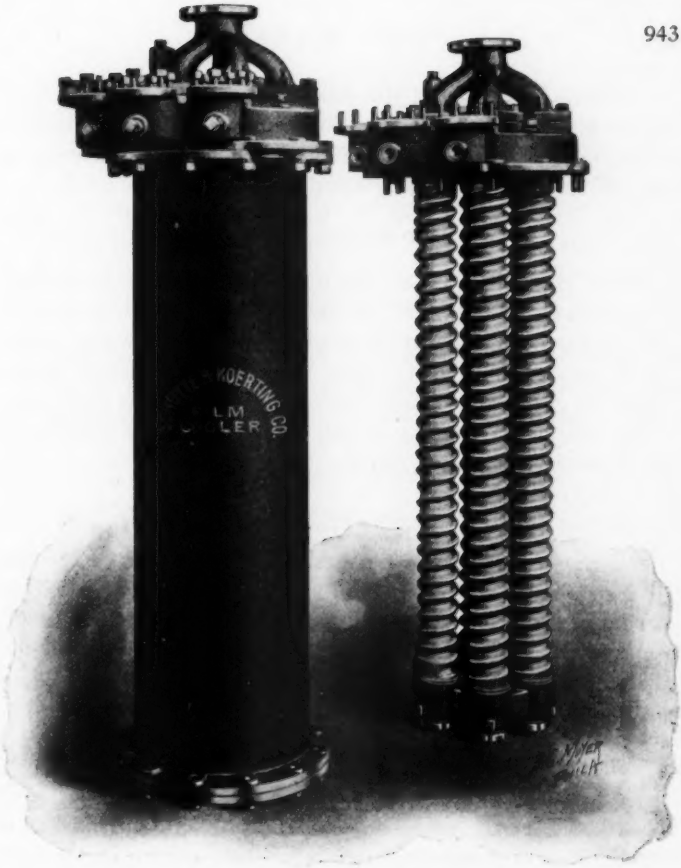
LUBRICATION SYSTEM.

The main turbine and line-shaft bearings are provided with a closed system of lubrication, the oil being forced through by two pumps provided especially for this service. These

pumps draw from tank located amidships at the lowest available place in the engine room, and force the oil through one or both, as the case may be, if required, of the coolers, from which it passes to the bearings at a pressure of about ten pounds per square inch, and is collected from the latter by a series of pipes that unite at a sight manifold provided with a thermometer for each bearing, so that the observer is able to note the temperature of oil passing from the bearings and the fact that there is a sufficient quantity as well. From this manifold the oil passes to the drain tank noted above and the operation is thus made continuous. Reserve tanks are located under the main deck in the engine room from which a supply of oil may be obtained to replenish that in other drain tank, should this be necessary on account of any leakage in the system. All piping is of copper or brass with composition fittings. For pumps and connections, see table of pumps.

The oil cooler used is the Schutte and Koerting corrugated-tube film cooler, consisting of six nests of concentric spirally-corrugated copper tubes, forming, by the differences in their diameters, "film spaces" through which the oil passes at a high velocity; the cooling water passing through the inside of the inner tubes and out through the space around the nest of tubes and the cooler shell. The tubes nest in "headers," that at the top being bolted to the shell flange and the bottom one free within the shell to permit of expansion. The cooling water and oil inlet and outlet are both at one end, this arrangement for the oil being accomplished by division walls in the headers to direct the flow. The shells are of copper, with composition flanges, heads, headers, and tube nuts. This cooler is shown on plate 1. The system will be recognized as that described in the article on Heat Transmission and Transmitter, by Captain A. B. Willits, U. S. N., in the February, 1910, number of this JOURNAL.

The advantages of this type of cooler, over the plain-tube type, are increased cooling surface per axial foot, self-cleaning on account of the constant agitation due to the tube corrugation, facility in replacing a tube or in removing same for cleaning.



Data for One Cooler.

Capacity per hour, gallons	2,000
From 150 degrees to 85 degrees, with water at 60 degrees.	
Length over all, feet and inches	4–04½
of shell, feet and inches	3–03
Diameter of shell, feet and inches	1-02
Number of tubes (nests)	6
Diameter of outer tube, O.D., inches	03½
inner tube, O.D., inches	02⅜
Gauge of tubes, Stubbs No	15
Length of outer tube, feet and inches	2–11
inner tube, feet and inches	3-06
Area of cooling surface, square feet	36

Constant agitation of the fluid to be cooled has the effect of bringing all of the particles in contact with the cooling surfaces, thus reducing to a minimum the size of the apparatus and the quantity of cooling agent used.

FEED-WATER HEATER.

There is one horizontal feed-water heater of the direct-flow type on the discharge side of the main feed pumps, located at the forward port side of the engine room. The water chests are of cast steel, shell of copper, tube sheets rolled naval brass, tubes 70 per cent. copper, 1 per cent. tin, 29 per cent. zinc. Total amount of heating surface measured on the outside of the tubes is 400 square feet.

FEED AND FILTER TANK.

There is one feed and filter tank located on the center line of the ship at the after engine room and as high as possible to permit of entrance to the filter chamber. Total capacity is 600 gallons, of which 235 is for the filter chambers and 365 for the feed tank proper. The tank is of steel, galvanized. There are the following connections to the first filter chamber:

Two main air-pump discharge, inches	9
Filling pipe, inches	2
Distiller fresh water, inch	$\frac{3}{4}$
Vapor pipe, inches	$1\frac{1}{2}$
Drain, inch	$\frac{3}{4}$

There are the following connections to the feed tank:

Main feed suction, inches	6
Overflow, inches	3
Vapor, inches	2
Drain, inches	$1\frac{1}{2}$

Water gauges and thermometer.

EVAPORATING AND DISTILLING PLANT.

There are two Williamson Brothers evaporators, total heating surface 136 square feet, located in the port after end of the engine room; and two Griscom-Spencer distillers, total cool-

ing surface 36 square feet, located on the after engine-room bulkhead near the evaporators.

The evaporators have a combined capacity of 2,600 gallons of potable water in twenty-four hours, and the distillers on test produced 396 gallons of potable water in four hours, or a rate of 2,376 gallons in twenty-four hours.

TURBO-GENERATING INSTALLATION.

The rated capacity of the dynamo is 5 kw., 125 volts. The turbine, which is of the Curtis type, is shown on page 823, Volume XXI, No. 3, of the JOURNAL, dated August, 1909.

TORSION METERS.

The shaft horsepower was calculated from readings furnished by Denny and Johnson's torsion meters. The installation is not a permanent fixture on the ship. One meter was applied to each line of shafting, located as well as possible under existing engine-room conditions. The torsional readings for the center line are taken from an actual length, less thickness of couplings, of 37 feet 3 inches, and that for the wing shafting being from a length of 23 feet 6 inches, less couplings. The recording boxes were located in the wireless-telegraph room on the main deck, which was well insulated in order to deaden all noise and vibration. Due to the delicacy of this apparatus two of the runs over the measured mile when standardizing had to be run over, on account of a slight disarrangement of the wires.

PIPING.

Space forbids a description of the various systems of piping, but plate No. 2 shows the general arrangement of the main and auxiliary steam pipes, at the same time furnishing a general idea of the arrangement of machinery in the engine and boiler rooms.

The following constitute the various systems of piping:

Main steam, auxiliary steam, receiver, auxiliary exhaust,

bleeder, dynamo steam and exhaust, main-feed suction and discharge, auxiliary-feed suction and discharge, bilge suction and discharge, sea suction and discharge, fresh-water suction from reserved-feed tanks, fire main, distilling plant, vapor and escape, heating system, steam to sea valves, air-compressor plant, lubrication system.

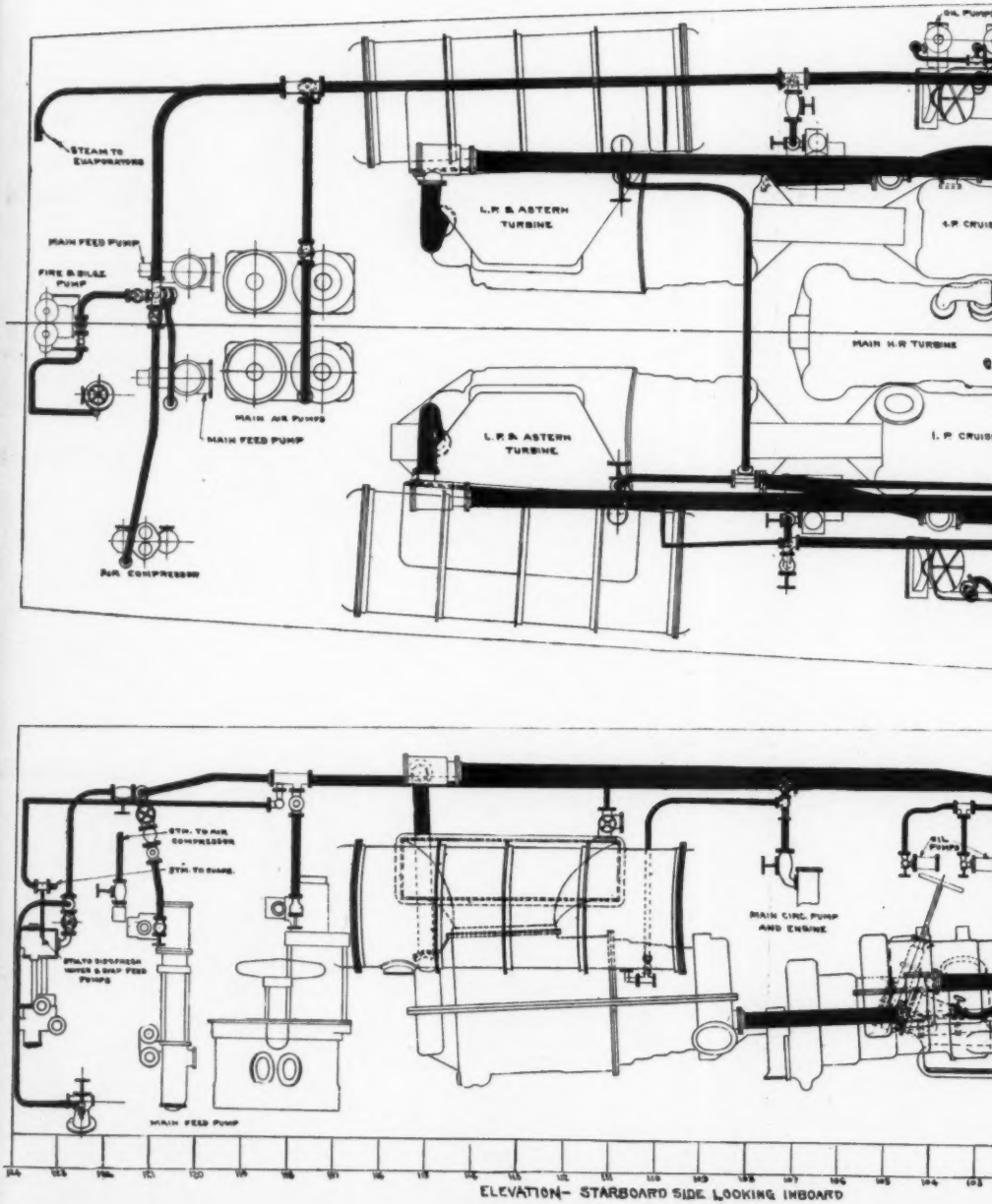
All steam pipes 2 inches and above are of seamless-drawn steel with rolled-steel flanges, and all below this are of copper with composition flanges. Main exhaust pipes are of sheet steel. All other exhaust pipes and feed pipes are of copper. Heater pipes are of brass. Drainage and pipes to reserve-feed tanks are of galvanized wrought iron.

OFFICIAL TRIALS.

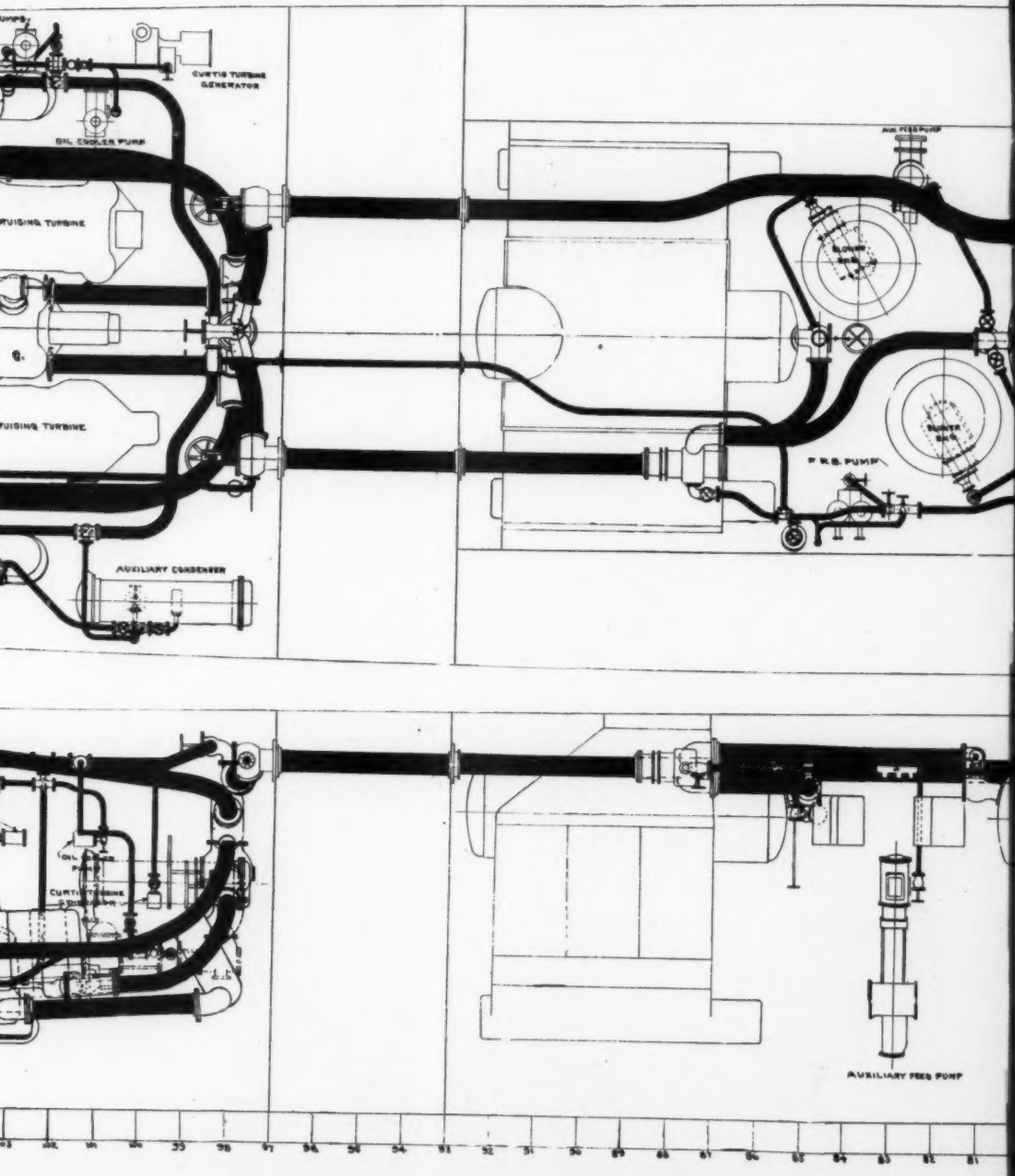
The Navy Department having authorized the trials to begin November 11th, 1909, the vessel left the shipyard at 9:18 A. M., reaching the Delaware Breakwater without incident at 2:40 P. M., time of run being 5 hours and 22 minutes, or at an average speed of 16 knots.

November 11th, the weather conditions being favorable, the standardization runs were begun at 9:30 A. M. and finished at 2:30 P. M. These runs were made over the measured mile inside the new breakwater, at the entrance to Delaware Bay, the maximum speed attained being at a rate of 31.364 knots, at 826.6 average revolutions, developing 12,018 S.H.P. for the main engines, and 12,504 horsepower for all machinery. The average for the five full-power runs was 30.67 knots at 836.22 average revolutions, developing 12,365 S.H.P., 12,838 horsepower for all machinery. The total horsepower is the sum of the shaft horsepower of main engines and the indicated horsepower of auxiliaries.

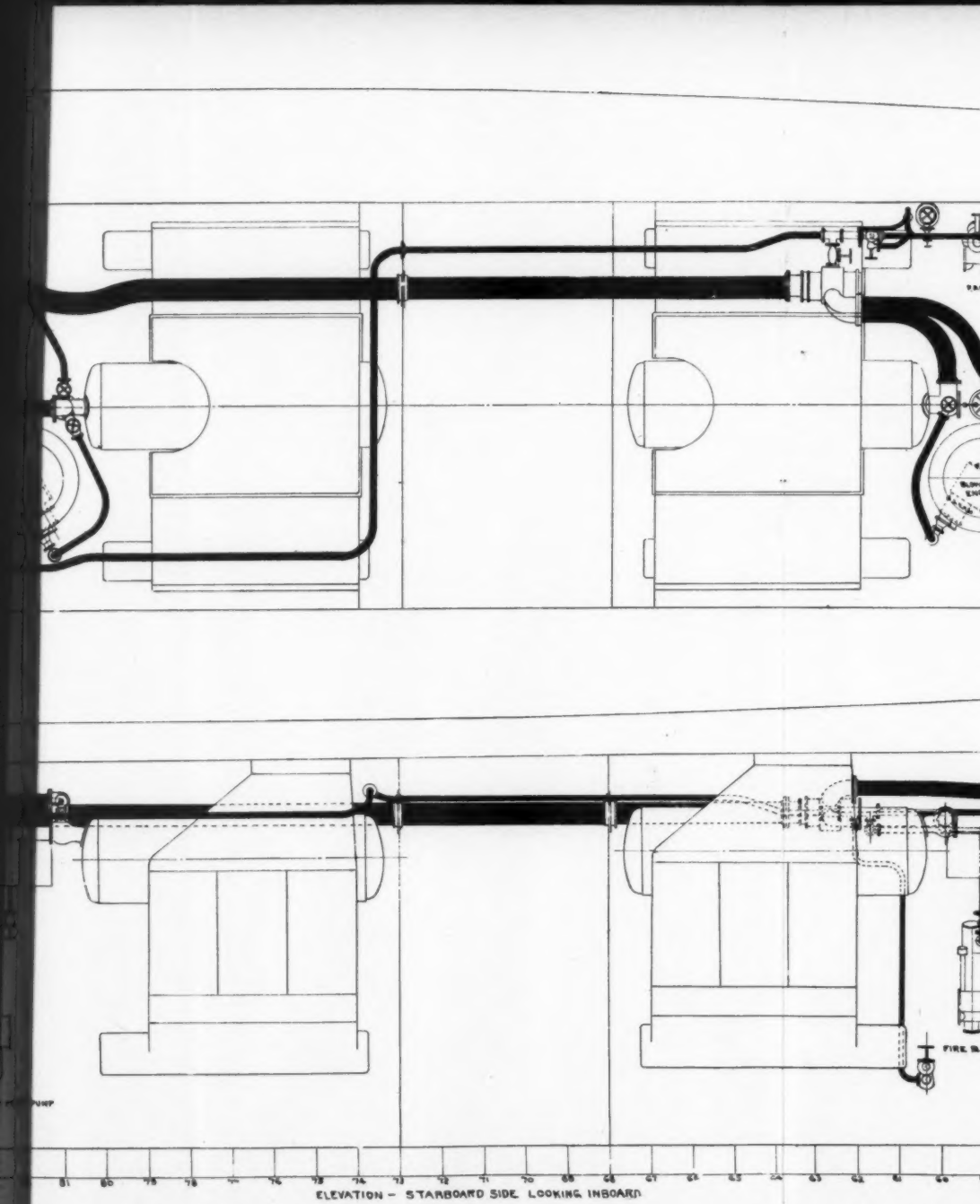
November 12th the four-hour full-speed trial in open sea was begun at 8:15 A. M., and concluded most successfully at 12:30 P. M. The weather conditions were perfect, sea being smooth, with a light N.W. wind and long easterly swell. At the conclusion of this run the usual backing tests were made between 2 and 2:30 o'clock P. M.



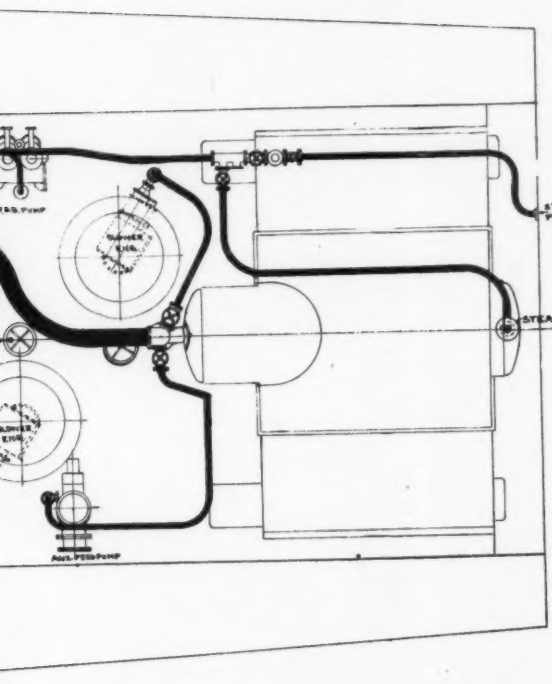
ELEVATION- STARBOARD SIDE LOOKING INBOARD



U. S. S. "PRESTON". ARRANGEMENT OF MACHINERY S



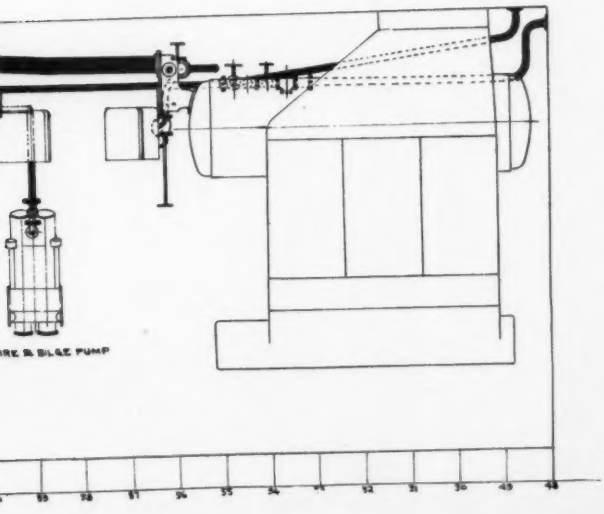
NERY SPACE, SHOWING STEAM PIPES.



U.S. TORPEDO BOAT DESTROYER

PRESTON

ARRANGEMENT OF MAIN STEAM AND AUXILIARY STEAM PIPES.



November 13th, the twelve-hour 24-knot run in open sea was begun at 7:30 A. M., and concluded most successfully at 7:30 P. M. The weather conditions were good, sea being smooth, light breeze varying from N.W. to N.E.

November 14th the twelve-hour 16-knot run in open sea was begun at 8:30 A. M., and was concluded most successfully at 8:30 P. M., after which, from 9 to 9:30 o'clock, the backing test on rudder was made. The weather conditions were good, sea conditions being light to moderate N.E. swell, with light to moderate N.E. breezes.

Results of Trials.

STANDARDIZATION RUNS.

For the purpose of standardizing the screws twenty runs were made over the measured-mile course. Previous to beginning these runs the draught was taken and found to be:

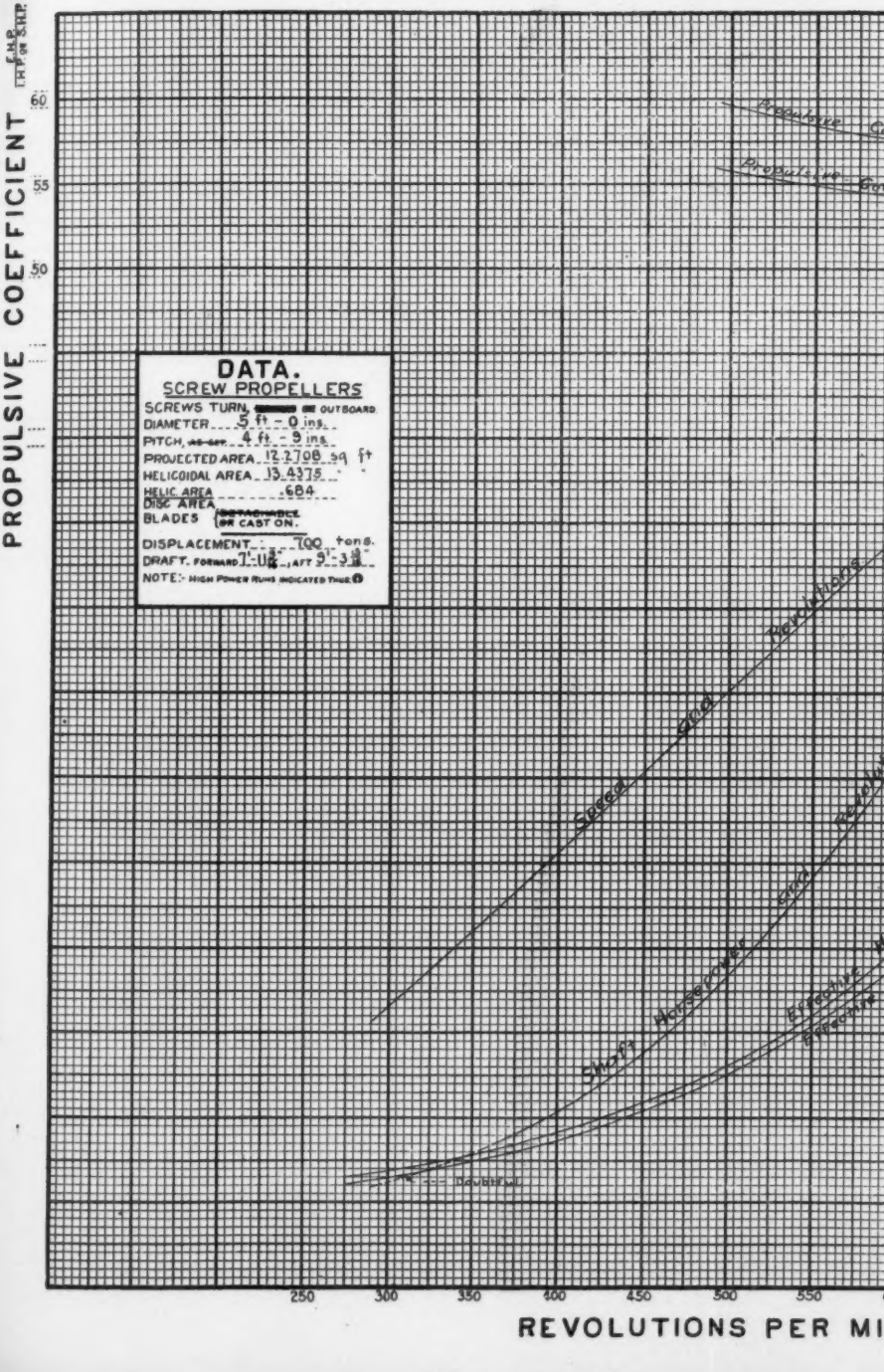
Forward, feet and inches	7-11⅛
Aft, feet and inches	8-01
Mean, feet and inches (even keel)	8-01
Corresponding displacement, tons	703

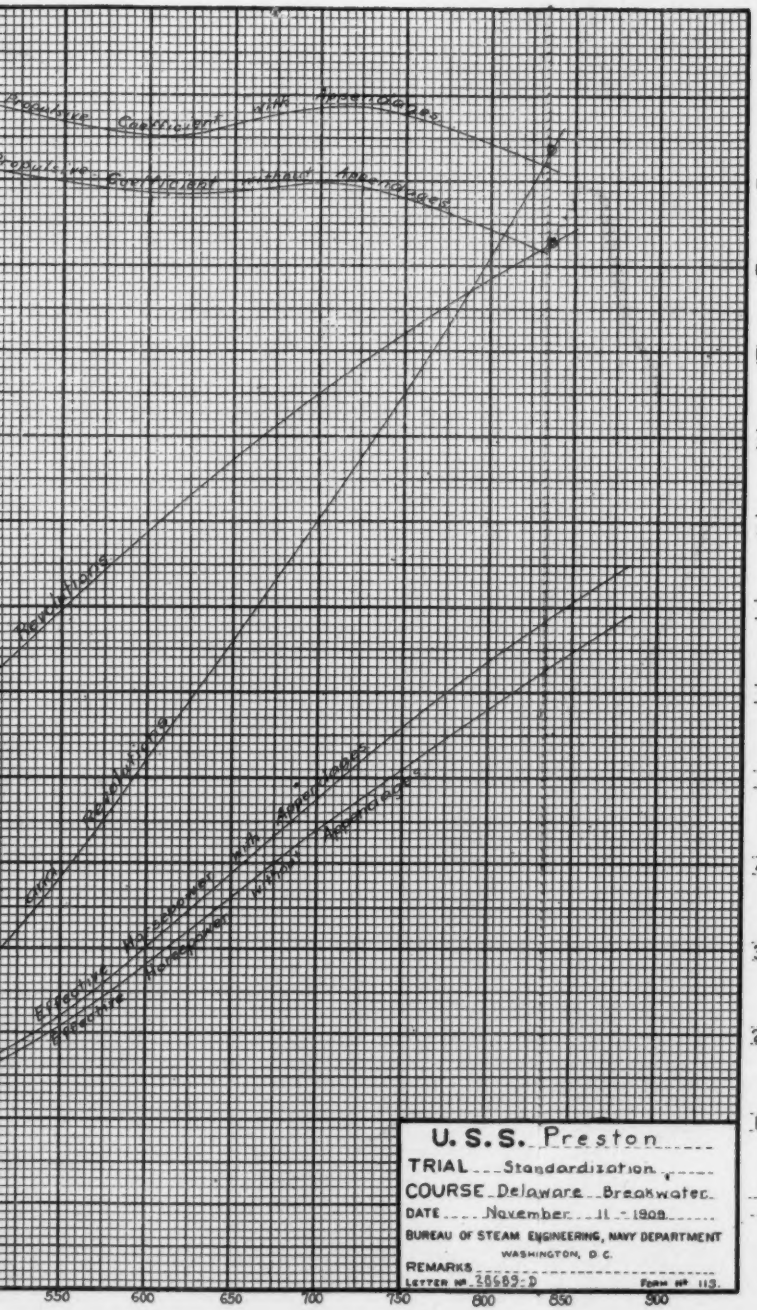
These runs give the data from which plate No. 3 "Standardization Curve" was plotted, and also that included in the table on page 949.

DATA FOR FOUR-HOUR RUN.

Mean draught on trial, feet and inches	8-02¼
Corresponding displacement at mean draught, tons	719
Trim by stern, inches	0-02½
Speed of ship, in knots per hour	29.177
Slip of propellers, in per cent. of their speed:	
Starboard	21.13
Center	18.30
Port	20.76
Mean slip	20.06

Revolutions, or double strokes, per minute (average of one-half-hour observations):		
Starboard shaft, average revolutions per minute		789.172
Center shaft, average revolutions per minute		761.868
Port shaft, average revolutions per minutes		785.527
Mean of all shafts, average revolutions per minute		778.855
	Starboard.	*Port.*
Pumps, main circulat'g, average revolutions per minute	341.0	333.0
air, average double strokes per minute	42.0	37.0
feed, average double strokes per minute	24.0	31.0
fireroom fire and bilge, average double strokes per minute	32.0	36.0
oil-cooler circulating, average double strokes per minute		53.0
Forced-draft blowers, average revolutions per minute		679.0
Forced-draft blowers, average air pressure, inches		2.84
Dynamo engine, average ampères		25.0
volts		121.0
Shaft horsepower:		
Starboard shaft		3,370.0
Center shaft		3,854.0
Port shaft		3,694.0
Collective of main engines		10,918.0
Indicated horsepower:		
Collective I.H.P. of all auxiliaries		438.0
Collective horsepower of main and auxiliary engines		11,356.0
Steam pressures (average of one-half-hour observations):		
Maximum at boilers, pounds		260.0
Mean at boilers, pounds		236.0
engine-room gauge, pounds		235.0
auxiliary exhaust, pounds		7.2
in steam belt M.H.P. turbine, pounds absolute		190.0
H.P.C. turbine, pounds absolute		7.4
I.P.C. turbine, pounds absolute		15.4
starboard L.P. turbine, pounds absolute		41.7
port L.P. turbine, pounds absolute		40.0
on glands, pounds absolute		1.9
Pressure in oil system, pounds		12.7
	Starboard.	*Port.*
Vacuum in condensers, inches of mercury, maximum	28.0	27.5
mean	27.5	27.2
Temperatures (average of one-half-hour observations):		
Injection, degrees	56.0	56.0
Discharge, degrees	85.0	93.0
Air-pump discharge, degrees		112.0
Feed water, degrees		178.0
Engine-room working platform, degrees		94.0


PROPULSIVE COEFFICIENT
60
55
50
DATA.
SCREW PROPELLERS
SCREWS TURN
OUTBOARD
DIAMETER 5 ft - 0 ins.
PITCH, as set 4 ft - 9 ins.
PROJECTED AREA 12.2708 sq ft
HELICOIDAL AREA 13.4375
HELIC AREA .684
DISC AREA
BLADES DETACHABLE OR CAST ON.
DISPLACEMENT 700 tons.
DRAFT. FORWARD 7'-11" AFT 9'-3"
NOTE:- HIGH POWER RUNS INDICATED THUS
Propulsive Coefficient
Shaft Horsepower
Effective Horsepower
250
300
350
400
450
500
550
REVOLUTIONS PER MI



PER MINUTE.

INDICATED, SHAFT AND EFFECTIVE HORSEPOWER.

SPEED IN KNOTS PER HOUR.

12000	32
11000	30
10000	28
9000	26
8000	24
7000	22
6000	20
5000	18
4000	16
3000	14
2000	12
1000	10
0	

Plate 3.

TURBINE DATA (ONE SHEET FOR ENGINE ROOMS) STANDARDIZATION TRIALS
(THREE-SHAFT ARRANGEMENT)

TURBINE DATA (ONE SHEET FOR ENGINE ROOMS) STANDARDIZATION TRIALS.

(THREE-SHAFT ARRANGEMENT.)

U. S. S. PRESTON, Course off DELAWARE BREAKWATER, on Nov. 11, 1909.

(FILL IN WITH INDIA INK FOR BLUEPRINTING.)

Trial (A-2948) Sheet 2.

NO. OF RUN.	TIME. H.	M.	S.	REVOLUTIONS PER MINUTE. SHAFT NO. 1.	SHAFT NO. 2.	SHAFT NO. 3.	AVERAGE ALL SHAFTS.	STEAM PRESSURES. ENGINE ROOM (GAGE). MAIN STEAM.	AUXILIARY STEAM.	AUXILIARY EXHAUST.	IN STEAM BELT (ABSOLUTE). MAIN H. P. TURBINE.	CRUISING TURBINES. H.P.	CRUISING TURBINES. I.P.	L. P. TURBINES. P.	L. P. TURBINES. S.	ON GLANDS (GAGE).	PRESSURE IN COLLING SYSTEM (GAGE).	VACUUM IN INCHES OF MERCURY. P.	VACUUM IN INCHES OF MERCURY. S.	TEMPERATURES. IN ENGINE ROOM.	INJECTION. P.	INJECTION. S.	DISCHARGE. P.	DISCHARGE. S.	AIR-PUMP DISCHARGE. P.	AIR-PUMP DISCHARGE. S.	HEATERS (FEED). FEED TO. P.	FEED TO. S.	FEED FROM. P.	FEED FROM. S.	OUTSIDE AIR.	TORSION METER READINGS. K=1.359 SHAFT NO. 1.	K=1.289 SHAFT NO. 2.	K=1.359 SHAFT NO. 3.	BRAKE HORSEPOWER. SHAFT NO. 1.	SHAFT NO. 2.	SHAFT NO. 3.	TOTAL.
1				308.2	283.94	313.57	301.84	214		8	14	100	43	5	6	1.5	7.5	28	27.8	72	54	54	72	80		89		109		210				.10					262	
2				310.31	285.20	314.56	303.36	230		7	14	101	42	5.5	5.5	1.5	8.0	28	27.8	72	55	54	73	77		88		109		216				.10					263	
3				314.69	289.03	315.35	306.36	218		7.5	145	100	42	6.5	5.5	1.5	7.0	28	27.8	76	54	54	74	81		89		109		214				.10					263	
4				413.00	373.61	415.39	400.83	235		7.5	28	176	81	8	8.0	2.0	8.0	28	27.8	92	55	54	74	80		89		104		216		.13		.17	449				591	
5				412.87	373.21	414.89	400.33	233		7.5	27	176	80	8	8.0	2.0	8.0	28	27.8	98	55	54	76	79		89		103		214		.13		.17	448				589	
6				413.64	374.16	415.88	401.22	240		7.5	28	176	81	8	8.0	2.0	8.0	28	27.8	98	55	54	77	79		88		104		216		.13	.165	.17	449	326	571	1346
7				644.10	584.81	570.19	593.03	236		7.	85	3	227	21	20.	3.0	11.0	28.	28.	94	56	54	82	88		84		96		194		.41	.45	.27	2206	1893	1284	4825
8				660.25	580.19	584.96	608.29	248		7.5	90	254	90	21	20.	4.0	11.5	28.	28.2	100	56	54	84	98		85		98		194		.42	.48	.28	2316	1978	1367	5164
9				662.50	573.11	572.56	601.39	240		7.6	90	253	233	20.	20.	2.5	11.0	28.	28.2	100	55	54	82	98		84		98		194		.43	.47	.29	2344	1939	1399	5167
10				678.29	682.21	679.61	680.03	228		7.	145	25	7.5	36	33	2.5	12.0	27.5	28.	100	54	55	90	94		84		101		190		.38	.43	.28	2153	2631	1390	6374
11				679.12	681.74	680.21	680.36	250		8.	147	25	12.5	30	33	2.5	13.0	27.7	27.8	100	56	56	88	94		85		102		184		.38	.42	.45	2156	2693	2557	7306
12				673.31	681.95	672.91	677.39	227		7.5	146	26	7.	29	33	2.5	13.0	27.5	27.5	94	56	56	90	98		87		102		190		.37	.44	.46	2091	2646	2607	7384
13				673.04	683.64	681.76	679.48	210		6.	145	25	7.5	29	33	2.0	12.0	27.6	27.2	96	54	54	90	100		86		100		180		.38	.44	.45	2133	2691	2563	7387
14				677.72	681.82	679.96	679.83	240		6.	145	0.0	7.5	29	33	2.0	13.0	27.5	27.8	94	54	54	86	94		84		100		188		.39	.44	.45	2208	2666	2566	7430
15				746.94	748.49	752.10	749.17	250		8.	185	25	7.5	32	935	2.0	14.0	27.3	27.3	96	56	56	88	98		86		109		188		.47	.50	.54	2986	3169	3419	9574
16				768.73	781.23	756.17	768.71	250		8.5	182	25	7.5	32	935	2.0	14.5	27.6	27.4	90	56	56	89	96		87		109		190		.47	.84	.54	3018	3338	3411	9763
17				756.79	746.42	764.09	764.43	247		6.	180	25	7.5	32	90	2.0	14.0	27.6	27.2	94	56	56	88	98		90		109		186		.46	.86	.53	2907	3372	3345	9644
18				841.22	806.63	830.06	826.64	260		7.	220	25	7.5	42	50	3.0	14.0	27.6	27.1	90	56	56	92	99		90		109		184		.54	1.00	.57	3794	4272	3952	12018
19				824.60	800.44	826.56	817.20	240		6.	210	25	7.5	46.	98	2.5	13.0	27.6	27.	92	56	56	92	102		92		100		178		.56	.99	.56	3788	4787	3866	11841
20				949.93	916.60	939.10	934.87	241		5.	215	25	7.5	42	99	2.0	12.0	27.5	27.	92	55	55	94	100		92		110		178		.57	1.04	.57	4047	4982	3995	12529
21				923.99	935.10	948.35	952.48	248		6.	223	25	7.5	48	51	2.0	12.0	27.5	27.	98	55	55	96	102		92		110		178		.52	1.04	.56	4141	4588	4064	12813
22				860.57	846.35	822.63	843.18	240		6.	218	25	7.5	46.	99	2.0	12.0	27.5	27.	96	55	55	94	100		94		110		176		.56	1.06	.56	4026	4740	3848	12614
23																																							
24																																							
25																																							
*TOTAL.																																							
*AVERAGE.																																							

Water consumption :	
Total amount used per hour, pounds	160,508.0
H.P. per hour, pounds	14.133
Note.—H.P. = S.H.P. of main engines plus I.H.P. of auxiliaries.	
Coal :	
Kind and quality used on trial.....Pocahontas, hand-picked, fair quality.	
Pounds used per hour, main and auxiliary engines	22,691.0
Knots run per ton of coal	2.88
Deduced data :	
S.H.P. per square foot of grate surface	31.20
heating surface	0.568
Pounds of coal per S.H.P. per hour	2.861
square foot of grate surface	64.84
Cooling surface (of main condensers) per S.H.P., sq. ft.	0.742

DATA FOR TWELVE-HOUR RUN AT 24 KNOTS.

Mean draught on trial, feet and inches	8–02	
Corresponding displacement at mean draught, tons	715.0	
Trim by stern, inches	0–02¼	
Speed of ship, in knots per hour	24.144	
Slip of propellers, in per cent. of their speed :		
Starboard	22.60	
Center	11.64	
Port	12.60	
Mean slip	15.61	
Revolutions, or double strokes, per minute (average of one-half-hour observations) :		
Starboard shaft, average r.p.m	665.37	
Center shaft, average r.p.m	582.79	
Port shaft, average r.p.m	589.19	
All shafts, average r.p.m	612.45	
	Starboard.	*Port.*
Pumps, main circulating, average r.p.m	263.0	264.0
air, average double strokes per minute	38.1	34.6
feed, average double strokes per minute	20.3	23.1
engine-room fire and bilge, average double strokes per minute	19.2	
fireroom fire and bilge, average double strokes per minute	32.1	21.4
oil-cooler circulating, average double strokes per minute	66.8	
oil-circulating, average double strokes per minute	47.2	
Forced-draft blowers (two only), average r.p.m	499.0	
average air pressure, inch	0.77	

Dynamo engine, average ampères		27.9
volts		120.5
Shaft horsepowers :		
Starboard shaft		2,497.0
Center shaft		1,566.0
Port shaft		1,486.0
Collective of main engines		5,549.0
Indicated horsepower :		
Collective I.H.P. of all auxiliaries		246.0
horsepower of main and auxiliary engines		5,795.0
Steam pressures (average of one-half-hour observation) :		
Maximum at boilers, pounds		260.0
Mean at boilers, pounds		254.0
engine-room gauge, pounds		249.0
auxiliary exhaust, pounds		5.2
in steam belt, I.P.C. turbine, pounds absolute		245.0
M.H.P. turbine, pounds absolute		95.1
H.P.C. turbine, pounds absolute		2.5
starboard L.P. turbine, pounds absolute		19.5
port L.P. turbine, pounds absolute		18.2
on glands, pounds absolute		1.7
Pressure in oil system, pounds		11.9
	Starboard.	*Port.*
Vacuum in condensers, inches of mercury, maximum	28.7	28.5
mean	28.4	28.3
Temperatures (average of one-half-hour observations) :		
Injection, degrees	56.0	56.0
Discharge, degrees	89.0	82.0
Air-pump discharge, degrees		91.0
Feed water, degrees		192.0
Engine-room working platform, degrees		97.0
Water consumption :		
Total amount used per hour, pounds		87,606.0
per H.P., pounds		15.118
Note.—H.P. = S.H.P. of main engines plus I.H.P. of auxiliaries.		
Coal :		
Kind and quality used on trial	Pocahontas, hand-picked, fair quality.	
Pounds used per hour, main and auxiliary engines		11,497.0
Knots run per ton of coal		4.7
Deduced data :		
S.H.P. per square foot of grate surface		15.854
heating surface		2.89
Pounds of coal per S.H.P. per hour		2.072
square foot of grate surface		32.849
Cooling surface (main condensers), square feet per S.H.P.		1.46

DATA FOR TWELVE-HOUR RUN AT 16 KNOTS.

Mean draught on trial, feet and inches		8–01
Corresponding displacement at mean draught, tons		704.0
Trim by stern, inches		0–9¾
Speed of ship, in knots per hour		16.135
Slip of propellers, in per cent. of their speed :		
Starboard		15.419
Center		5.358
Port		16.481
Mean slip		12.419
Revolutions, or double strokes, per minute (average of one-half-hour observations) :		
Starboard shaft, average r.p.m.		406.85
Center shaft, average r.p.m.		363.60
Port shaft, average r.p.m.		412.02
All shafts, average r.p.m.		394.15
	Starboard.	*Port.*
Pumps, main circulating, average r.p.m.	272.6	246.3
air, average double strokes per minute	33.8	31.0
feed, average double strokes per minute	13.3	...
engine-room fire and bilge, average double strokes per minute		19.5
fireroom fire and bilge, average double strokes per minute	36.0	18.0
oil-cooler circulating, average double strokes per minute		41.2
oil circulating, average double strokes per minute.		40.0
Forced-draft blowers (one only), average r.p.m.		363.0
average air pressure, inch		0.4
Dynamo engines, average ampères		34.5
volts		121.0
Shaft horsepowers :		
Starboard shaft		457.0
Center shaft		309.0
Port shaft		590.0
Collective of main engines		1,356.0
Indicated horsepower :		
Collective I.H.P. of all auxiliaries		146.0
horsepower of main and auxiliary machinery		1,502.0
Steam pressures (average of one-half-hour observations) :		
Maximum at boilers, pounds		248.0
Mean at boilers, pounds		219.0
engine-room gauge, pounds		219.0
auxiliary exhaust, pounds		5.38
in steam belt, H.P.C. turbine, pounds absolute		173.0
I.P.C. turbine, pounds absolute		77.0
M.H.P. turbine, pounds absolute		28.6

Steam pressures (continued):		
Mean in steam belt, starboard L.P. turbine, pounds absolute		7.11
port L.P. turbine, pounds absolute		6.65
on glands, pounds absolute		1.59
Pressure in oil system, pounds		9.5
	Starboard.	*Port.*
Vacuum in condensers, inches of mercury, maximum	29.0	29.0
mean	28.5	28.6
Temperatures (average of one-half-hour observations):		
Injection, degrees	55.7	55.8
Discharge, degrees	76.5	74.3
Air-pump discharge, degrees		73.1
Feed water, degrees		209.0
Engine-room working platform, degrees		88.7
Water consumption:		
Total amount used per hour, pounds		32,219.0
per H.P., pounds		21.451
Note. —H.P.= S.HP. of main engines plus I.H.P. of auxiliaries.		
Coal:		
Kind and quality used on trial	Pocahontas, hand-picked, fair quality.	
Pounds used per hour, main and auxiliary engines		3,879.0
Knots run per ton of coal		9.316
Deduced data:		
S.H.P. per square foot of grate surface		7.749
heating surface		0.141
Pounds of coal per S.H.P. per hour		2.861
square foot of grate surface		22.166
Cooling surface (main condenser) per S.H.P., sq. feet		5.975

INFLUENCE OF DEPTH OF WATER ON SPEED, REVOLUTIONS AND HORSEPOWER.

SPECIAL TRIALS OF "MICHIGAN," "FLUSSER" AND "REID."

By Lieutenant Commander U. T. Holmes, U. S. N., Member.

In order to determine the relative value of the three trial courses at Rockland, Me., Provincetown, Mass., and Delaware Breakwater, and to obtain data as to the influence of depth of water on a vessel's performance, tests were made, under the direction of the Navy Department, during the latter part of May of this year.

The following table gives the particulars of these courses as regards depth of water:

.....	Length of measured course, miles.	Depth on course, fathoms.		Depth on approaches one, mile from course, fathoms.	
		Least.	Greatest.	Least.	Greatest.
Rockland	1.0	53.0	79	40	79
Provincetown.....	1.00013	25.0	27	22	28
Delaware Breakwater..	1.2415	22.5	25	16	25

The Rockland course is practically free from cross currents. A tidal cross current is sometimes found on the Provincetown course, for which corrections are made. The influence of the tidal current in and out of Delaware Bay is felt at the Delaware Breakwater course, making eddies that render it difficult to apply tidal observations and obtain absolutely correct results on this course.

The battleship *Michigan* and destroyers *Flusser* and *Reid* were used for the trials. All vessels were docked and their bottoms and propellers cleaned early in May. The conditions as to draught and displacement were kept as nearly as

possible uniform on all the trials. The *Flusser* and *Reid* are built from the same plans and are similar in all respects, so that each should furnish a check on the other.

Full particulars concerning the machinery installations of these vessels have been published in the JOURNAL, as follows: *Michigan*, Vol. XXI, No. 3, August, 1909; *Flusser* and *Reid*, Vol. XXII, No. 1, February, 1910. The general particulars, which are of interest in connection with these trials, are as follows:

	Michigan.	*Flusser.*	*Reid.*
Length over all, ft. and ins.....	450-00	293-10½	293-10½
Beam, feet and inches.....	80-02¾	26-05	26-05
Draught, forward, ft. and ins..	23-09¾	7-06½	7-04
aft, ft. and ins.....	25-01½	8-06½	8-08
mean, ft. and ins.....	24-05¾	8-00½	8-00
Displacement, tons.....	15,998.5	692.5	691
Propellers:			
Number.....	2	3	3
Type.....	3-blade mang. bronze, built up, machined.	3-blade mang. bronze, solid, machined.	3-blade mang. bronze, solid, machined.
Diameter, feet and inches...	17-06½	5-03	5-03
Pitch, feet and inches.....	18-00	4-10	4-10
Developed area, sq. ft.....	88.5	14.778	14.778
Projected area, sq. ft.....	75.28	12.98	12.98

REMARKS.

For the *Michigan* the revolutions were practically the same on all the courses, for all speeds. The influence of shallow water on the horsepower is felt at 18 knots, and at 19 knots there is a wide difference in favor of the deep-water course.

For the destroyers the revolutions are approximately the same on all the courses for speeds up to about 27 knots, when the deep-water course begins to show a marked advantage. At speeds between 31 and 32 knots the Rockland course gives half a knot more for the same revolutions. A marked decrease in power for the same speed in deep water begins to make itself manifest, in the case of the destroyers, at about 25 knots' speed. At the highest speed the Rockland course is about one knot faster than the Delaware Breakwater course.

Minor corrections in the data may be made when full reports of the trials are received. The general conclusions as given will not be changed.

Trial Data.

.....	12 knots.		15 knots.		18 knots.		19 knots.	
Michigan	*R.p.m.*	*I.H.P.*	*R.p.m.*	*I. H.P.*	*R.p.m.*	*I.H.P.*	*R.p.m.*	*I.H.P.*
Rockland	72.1	3,300	91.1	6,690	113.0	13,325	121.6	16,880
Provincetown	71.3	3,360	90.2	6,610	113.4	13,490	121.8	17,175
Delaware Breakwater	72.5	3,525	91.2	6,900	113.5	13,525	122.9	18,300

.....	17 knots.		20 knots.		22 knots.		25 knots.		28 knots.		30 knots.		31 knots.		32 knots.	
Flusser	*R.p.m.*	*S H.P.*	*R.p.m*	*S.H.P.*	*R.p.m.*	*S H P.*	*R.p.m.*	*S.H.P.*	*R.p.m.*	*S.H.P.*	*R.p.m.*	*S.H.P.*	*R.p.m.*	*S.H.P.*	*R.p.m.*	*S.H.P.*
Rockland	393	1,425	464	2,510	520	3,550	608	5,700	701	8,700	779	11,110	821	12,400	863	13,725
Provincetown	388	1,590	461	2,600	518	3,730	604	6,200	701	9,330	786	11,860	831	13,230	877	14,710
Delaware Breakwater ..	396	1,685	466	2,600	522	3,685	606	6,040	705	9,200	795	12,200	841	13,600	886	14,980
Reid																
Rockland	394	1,325	466	2,220	523	3,310	609	5,425	697	8,200	774	10,610	818	11,890	...	...
Provincetown	392	1,185	466	2,080	523	3,090	612	5,470	709	8,560	791	11,050	835	12,400	...	...
Delaware Breakwater ..	396	1,330	467	2,225	524	3,285	608	5,530	707	8,585	789	11,275	844	13,000	...	...

NOTES ON THE OPERATION AND MANAGEMENT OF THE CURTIS TURBINES ON THE U. S. S. *SALEM.*

By Lieutenant W. W. Smith, U. S. N., Member.

Note.—Lieutenant Smith's experience as Senior Engineer Officer of the *Salem* enables him to write advisedly on the care and operation of machines of this type.—Editor.

These notes have been prepared with the object of placing before the readers of the Journal a brief guide for the operation and management of Curtis marine turbines, so that those who follow will profit by the experience gained and lessons learned, with this type of machinery in service.

When the *Salem* was commissioned the officers and men had had practically no experience with such machinery, and the lack of practical information was keenly felt. In the future it is possible that others may be placed in the same position, and will feel the need of such information as is contained in these notes.

It is hoped that these notes, though incomplete, may form the groundwork for discussion, and that they may be added to by those who gain further experience.

Reference is made only to the main turbines and the turbine auxiliaries of new type. The notes apply particularly to the turbine machinery of the *Salem,* which differs from subsequent installations of the Curtis type as noted below. Reference is invited to a description of this machinery by Mr. Chas. B. Edwards (J. A. S. N. E., Vol. XX, No. 4, November, 1908). The following particulars should be noted:

Both turbines are right-hand, the starboard facing forward and the port facing aft. The foundations for thrust bearings are built up on the structural part of the ship and are not

connected to the turbine casings except by braces. In the starboard turbine the unbalanced steam pressure on the turbine blading is opposed to the thrust of the ship both ahead and in backing, and consequently there is little load on this thrust bearing. In the port turbine the thrust of the turbine and ship act in the same direction, and consequently the port thrust carries a double load. The operating valves are balanced by steam pistons, and are operated by levers. Oil cylinders are fitted to regulate the control. When the rotor is lifted, it is also necessary to lift the diaphragms. Clearance gauges for turbine rotors are fitted at each end.

The condensing plant is of the dry-air type, as furnished by the International Pump Co. The condenser is fitted with baffles in the lower part to separate air and water. There are two air pumps for each condenser—one for air and one for water. The dry air, or "dry-vacuum," pump is of the rotary type. The wet air, or "hot-well," pump is a two-stage centrifugal pump driven by a two-stage Curtis turbine. Both pumps are designed to run at constant speed and are regulated by governors.

The turbine bearings are fitted with forced lubrication of the closed type, not cross connected. The cooling tank is on the return side.

The propellers are of the four-bladed solid type, machined true.

Subsequent Curtis turbine installations differ from that on the *Salem* principally as follows:

The turbines are right and left-hand, both facing forward. The propeller thrust is balanced by the turbine thrust, and, in consequence, the loads on thrust bearings are small. The thrust bearings are reduced in size and are rigidly connected to turbine casings. The main bearings are connected to turbine casings and are not self-aligning. These differences remove most of the difficulties herein noted on account of lack of clearance or improper adjustment of clearance. Later installations are in this respect superior to the starboard turbine on the *Salem*. Other improvements are as follows:

The exhaust pipe is of rectangular section, built up of steel plates. The condensers are located abaft the turbines. The casing construction is all of cast iron and in fewer sections. The operating valves are of the screw-operated type. The diaphragms are made of steel plates. The blading is of improved construction. The air pumps are of different type. This, however, is a difference, and not necessarily an improvement, since the condensing apparatus on the *Salem* has been very satisfactory.

In referring to clearance, the terms may be defined as follows:

Radial Clearance: The space between rotor and stator in a radial direction. In most cases it may be defined as the vertical space between shaft and diaphragm bushings or between the tips of buckets and casing.

Longitudinal Clearance: The space between rotor and stator in an axial, or fore-and-aft direction.

Ahead Clearance: The space between rotor and stator in the ahead direction, or the distance the rotor would have to move in the ahead direction before striking.

Astern Clearance: The reverse of above; the clearance space abaft the rotor.

Total Clearance: The sum of the ahead and astern clearances, or the total distance the rotor can be moved between the ahead and the astern striking points.

Float: The amount of play in the thrust bearing.

Change of Clearance: The amount of play in the rotor, or the amount that the clearance changes.

In this type of turbine, it should be remembered that it is necessary to be informed of the clearances on both ends of the rotor.

OPERATION.

The question of operation is taken up under the following headings:

Warming Up; Starting; Maneuvering; Underway; Standing by; Full Power; Shutting Down.

WARMING UP—NORMAL CONDITIONS.

Time Required.—About three hours should be allowed for warming up under ordinary circumstances. A shorter time may be allowed if the turbines have been running and are already partially warmed. Under such circumstances the time allowed will depend on the speed to be obtained and the heat already in the turbines.

As a general rule, the turbine should be worked up to speed gradually, especially for high speeds, so that all parts will expand uniformly.

Starting Cold—Emergency Conditions.—On the *Salem* it was not found to be good practice to start the turbines cold, and it should never be done except in extraordinary emergency. On account of the narrow margin of clearance, the port turbine is likely to rub in backing if it is not properly warmed. The starboard turbine can, if necessary, be started cold without difficulty, because there is ample margin in the clearance. In case of emergency, when it will not be necessary to back, and when the port turbine can be stopped or slowed down, both turbines can be started as soon as steam is raised, and worked up to speed as fast as is practicable without rubbing. The port turbine will require a longer time to work up to speed than the starboard turbine.

If the turbines are started before they are properly warmed up leaks are likely to start in the joints, which will require considerable work to repair.

Precautions.—All parts of the turbine should be warmed uniformly. Steam should be admitted for this purpose at each end, and the by-passes of the ahead and astern valves cracked open to warm the piping and steam chests.

The rotor should be frequently turned with the turning engine while the turbines are being warmed, so that it will heat up uniformly. A good plan is to move it a quarter of a turn every ten or fifteen minutes.

It should be remembered that the turbine can be only partially warmed in this way, and that the normal running heat

is not attained until the turbine is actually running with the stage pressures corresponding to the speed. Consequently, it is necessary to work up gradually to the running heat after getting underway.

The clearances should be taken before and after warming up, and noted in the log. The minimum clearance under these conditions should be about 0.05 inch.

Routine.—The following routine should be followed: Start main circulating and hot-well pumps, observing the usual precautions in warming up and starting.

Take clearances and note in log. See that everything is clear for turning the turbine.

Open warming steam to both ends of the turbines. Open drains to the turbine stages and start siphon to clear the turbine of water.

Turn on steam seals of turbine glands, crack drain valves, and adjust pressure to 3 or 4 pounds above atmosphere.

Run oil pump for several strokes to flood the bearings.

Every ten or fifteen minutes turn the rotor about a quarter of a turn so that it will be warmed uniformly.

Fill the oil-control cylinders of the operating valves, and see that the connecting valve is properly adjusted.

Drain water from oil tank and see that it is filled to the top of the glass.

See that thrust bearings contain sufficient lubricant—a mixture of engine oil and water. (This applies only to thrust bearings of the open type as fitted on the *Salem.*)

Examine oil service and see that the proper valves are open. Examine oil strainer and see that it is clear.

Test water service and see that the water is flowing properly.

When steam is on the main line, crack the by-passes of the ahead and astern operating valves so as to warm up piping and steam chests.

Open required number of nozzles on ahead steam chest. See that all gauges are connected and working properly. See that the separator trap is working properly.

About half an hour before geting underway disconnect turning gear, and see that everything is clear.

Start dry-vacuum pump, making sure that water is circulating through the jacket of the air cylinder and that the cylinder and valves are properly lubricated.

Start oil pump and regulate oil pressure to about 30 pounds.

In trying the turbine, open steam to the ahead end just enough to move it several revolutions, then shut off ahead and do the same with the reverse end.

Shut down oil pump to prevent loss of oil. If this is not done the oil will escape past the centrifugal throws to bilge.

Report ready, and stand by to answer signals at the time set for getting underway.

Take clearance for the second time and note in log.

Circulating Pump.—Should be started before warming steam is turned on turbines. The usual precautions are observed in warming up and starting.

Hot-Well Pump.—A section through this pump is shown in Fig. 1. It should be started after the circulating pump is running. Open suction and discharge valves and valve to water seal of pump. Fill oil cup, flood bearings, and oil governor.

See that the proper number of steam nozzles are open. Open exhaust to condenser and drains from glands. Open steam valve and bring up speed slowly. Adjust steam seals of glands to prevent leakage of air.

The exhaust should always be opened to main condenser, so that the turbine will benefit by a high vacuum and run under the most economical conditions. If the exhaust is opened to the auxiliary exhaust line the turbine will not run economically.

Two nozzles are used for speeds up to 15 knots, and three nozzles for greater speeds.

Care should be taken that the water seal of the pump is working properly, otherwise the vacuum will be impaired.

In adjusting the steam seals the drain valves should be cracked sufficiently to drain off the water. They should not be opened too wide, otherwise steam will be wasted. It is not necessary to start the hot-well pump until just before

getting underway, but it is better to start it when beginning to warm up, as the condenser is drained better in this way. If too much water collects in the condenser the valves of the dry-vacuum pump may be damaged by water.

The clearance of the turbine should be noted. This is not very important, as the clearance has not changed nor been adjusted in two years.

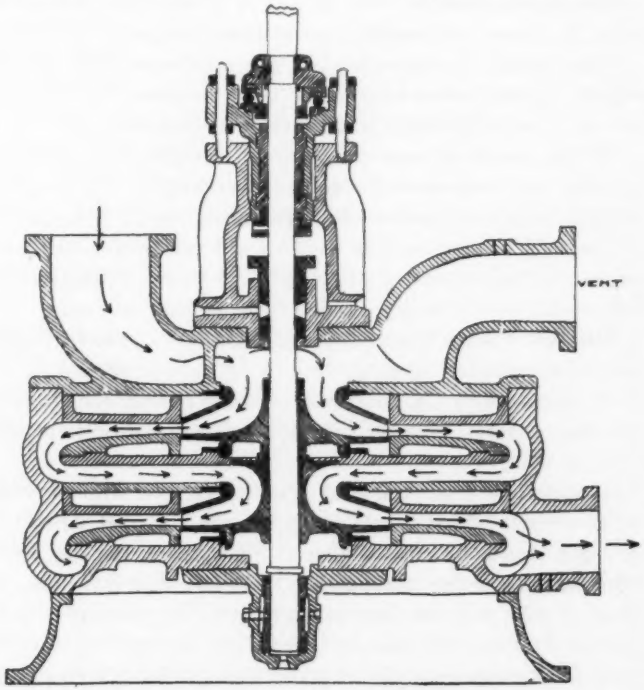


Fig. 1.—CENTRIFUGAL HOT-WELL PUMP.

The revolutions per minute should be taken, and should be about 1,250. If necessary, the governor should be adjusted to give the required speed.

Dry-Vacuum Pump.—Open exhaust and all drains—steam cylinder, air cylinder, air valve chest, and oil separator. Start

lubrication and oil governor. See that the air-cylinder lubricator is filled with special cylinder oil—valvoline or arctic oil. See that water is circulating through jacket. Adjust valves to an early release.

Stand by steam valve and suction valve to condenser. Turn by hand with jacking bar, and open steam and condenser suction.

When a vacuum of over 20 inches is produced, adjust the valve to about mid position or medium release.

Care should be taken to see the condenser well drained before starting, otherwise water will be drawn into the suction and the air cylinder and valves may be damaged.

If the water is not circulating through the jacket the cylinder and valves will become overheated. Only special cylinder oil should be used for lubricating the air cylinder and valves. If ordinary oil is used it will carbonize, burn, and produce a large quantity of smoke. If its use is continued it will injure the valves by causing them to heat and cut.

The best results have been obtained with a speed of about 55 revolutions per minute, and the governor should be adjusted accordingly. Care should be taken to keep the governor running properly, because this pump will not run satisfactorily without the governor.

Considerable trouble was at first experienced with the chain on governor, which would jump off the sprocket wheel. A spring was attached to the idler to give greater tension, which effectually remedied the difficulty. Before this was done, the pump would stop frequently while running, causing a great deal of trouble. No danger is involved through its stopping, since the condenser will carry the vacuum for a considerable interval, and the hot-well pump alone will maintain a vacuum of about 10 inches.

Oil Service.—When starting to warm up, the oil pump should be run for a few strokes to flood the bearings, but it should not be kept running, since the oil will leak by the centrifugal throws to bilge. It should be started again when turning the turbines and shut down until ready for getting

underway. When the turbines are running the centrifugal throws prevent the loss of oil.

The oil service should be gone over carefully to see that the proper valves are open. *The valves of vital importance are the ones to the lower brasses. If these are not open the bearings are sure to run hot.*

The oil strainer should be removed, examined, and cleaned if necessary. It should always be used and should never be by-passed except when it is being examined.

Water collects in the bottom of the oil tank, and this should be drained off, but care should be taken not to drain off any of the oil, and to have the drain valve tightly closed.

The oil tank should be filled to the top of glass before starting. The level will fall to about three-fourths when the system is filled in running, but it should not be carried higher than this, because it will overflow and be wasted in case of stopping, when the oil in the system drains back into the tank.

In filling the tank, care should be taken to fill with *turbine* oil and not *engine* oil. Ordinary engine oil is not suitable for forced lubrication on account of its tendency to emulsify.

See that water is circulating through the cooling coils of the oil tank.

See that the thrust bearings contain sufficient lubricant—a mixture of half engine oil and half water. In later vessels the thrust bearings are fitted for forced lubrication and receive attention similar to that given other bearings.

Water Service.—When the circulating pump is started, open discharge and suction valves to the water service. See that all necessary valves are open and that water is circulating properly. Use test cocks, and see that none of the pipes or jackets are air bound. The following should be examined and tested: Main bearings, thrust bearings, cooling coil of oil tank and jacket of air cylinder of dry-vacuum pump.

Operating Valves.—See that oil-control cylinders are filled and that the connecting valve is properly adjusted. See that

the operating levers and gear are properly adjusted, so that the valves seat properly.

With the type of operating valves fitted on the *Salem,* the steam pressures acting on the valve do not balance properly when opening, and, if the oil-control cylinders are not properly filled and adjusted, the valve opens violently, thus throwing the operating lever through the extreme throw. Later vessels fitted with Curtis turbines have operating valves of the screw-down type.

Drains.—When starting to warm up, open all drain valves between stages and the discharge valve to the condenser. Then open the steam to the siphon to expel the water. Keep the drains open and steam on the siphon until the turbine has settled down to running conditions.

In closing the drains, close all valves mentioned above, and see that the gland drains are open to the fourth stage.

It has not been found necessary to use the drains while underway, as the water is entrained and carried to the condenser.

Glands.—Steam should be turned on and the glands sealed when starting to warm up. This is not necessary, but it is desirable to prevent cold air from entering the turbine.

The gland pressure should be just sufficient to seal—slightly above atmosphere, about 3 or 4 pounds. The drain valves should be cracked sufficiently to take care of the condensation, but should not be opened wide.

STARTING.

Just before getting underway, or when a signal is received, start the oil pump and maintain a pressure of about 30 pounds in oil service.

Increase the speed of the circulating pump and note the discharge-water temperature, particularly in backing.

See that the steam seals are properly adjusted and that a vacuum of 28 inches or more is being maintained.

Examine sight boxes and see that oil is flowing through

properly. Go over oil and water service and all bearings to see that they are running properly.

See that warming steam is shut off tightly, and that the astern operating valve, balance steam, and by-pass are tightly closed.

After the turbine has settled down to running conditions close drains and shut steam off of the siphon.

Clearance.—Note the clearance frequently, and, if it is too small, it is probable that the turbine has been worked up to speed too quickly, and it should be slowed down and worked up more slowly to permit uniform expansion.

Auxiliary Exhaust.—After settling down to running conditions the auxiliary exhaust should be opened into the third or fourth stage, depending on the stage pressures. It is nearly always opened into the third stage, for it is only at very high powers that the pressure on this stage exceeds the auxiliary exhaust pressure.

MANEUVERING.

In maneuvering at any time the following should be observed:

Operating Valves.—Before moving the main operating valve open the balance steam and the by-pass. In opening the main valve open it gradually and do not throw it open suddenly. If this is not done the unbalanced pressure may cause the valve to "take charge" and result in an accident.

Clearance.—The greatest change of clearance is when the turbine is reversed, and consequently the clearance should be noted when this is done.

In the case of the starboard turbine on the *Salem,* where the thrust of the propeller and the steam pressure act in opposite directions, there is only a small load on the thrust bearing. The clearance of this turbine gives very little trouble for this reason. In the extreme condition of reversing at full speed the change of clearance is less than 0.05 inch. This is the condition that will obtain in both turbines in later installations.

In the case of the port turbine, where the thrust of the

propeller and the thrust of the turbine act in the same direction, it is deflected forward in going ahead and aft in backing. As the thrust bearing fixes the fore-and-aft position of the rotor in relation to the stator, this deflection causes the rotor to change its fore-and-aft position, resulting in an excessive "change of clearance" in reversing.

It should be remembered that there are three causes for the change in clearance, and that they vary in importance according to the speed: (1) The "float" of the thrust bearing, or the clearance which is allowed for lubrication; (2) the difference of expansion between the rotor and the stator; (3) the deflection of the thrust bearing in relation to the stator.

In the port turbine on the *Salem* the maximum values for these are (1) 0.018, (2) 0.06, (3) 0.023. It is not safe to allow less than 0.02 inch clearance in running. To assure safety about 0.05 inch is allowed. As the change of clearance is greater than the total clearance, it is not possible to back with the turbine cold and run full speed ahead with the turbine at its running heat on the same adjustment. Consequently it is necessary to adjust the clearance between these conditions to prevent rubbing.

As most running on the *Salem* is done at speeds below 20 knots, the clearance should be adjusted for the lower speeds. See clearance curves, Figs. 2, 3 and 4.

Revolutions.—The revolutions can be roughly determined by the number of nozzles open and the steam-chest pressure, but this is not very satisfactory. Speed indicators should be used for this purpose.

Backing.—It should be remembered that the backing turbines use about three times as much steam for the same power as the ahead turbines, and consequently the speed of the forced-draft blowers should be increased if backing is continued. The drop in steam pressure is considerable when there are only a few boilers steaming, and for this reason it is advisable to have a good head of steam for maneuvering. It is not safe to allow the pressure to fall much below 150 pounds, on account of the difficulty in keeping the auxiliaries up to speed.

The best results have been obtained by carrying the pressure at about 250 pounds before backing and not allowing it to fall below 150 pounds while backing.

When all boilers are connected, the full backing power can be obtained easily with but small loss of pressure. (See "Backing Tests of Scout Cruisers," p. 831.)

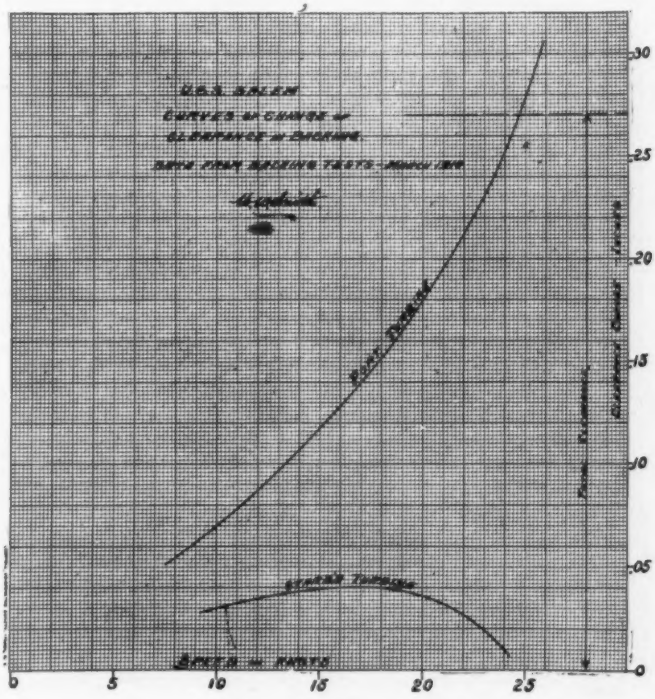


Fig. 2.

With a small number of boilers in use the pressure frequently drops 100 pounds in about a minute when backing, but this has never caused any noticeable priming or other trouble.

Auxiliary Exhaust.—When maneuvering is expected, the auxiliary exhaust should be opened into the condenser and

shut off from the turbine. However, no harm will result if this is not done, as under such conditions the turbine can be reversed without inconvenient results.

Nozzles.—When maneuvering is expected, additional nozzles (on the ahead end) should be opened, so that reserve power can be developed without delay.

On the backing end fourteen nozzles are always open;

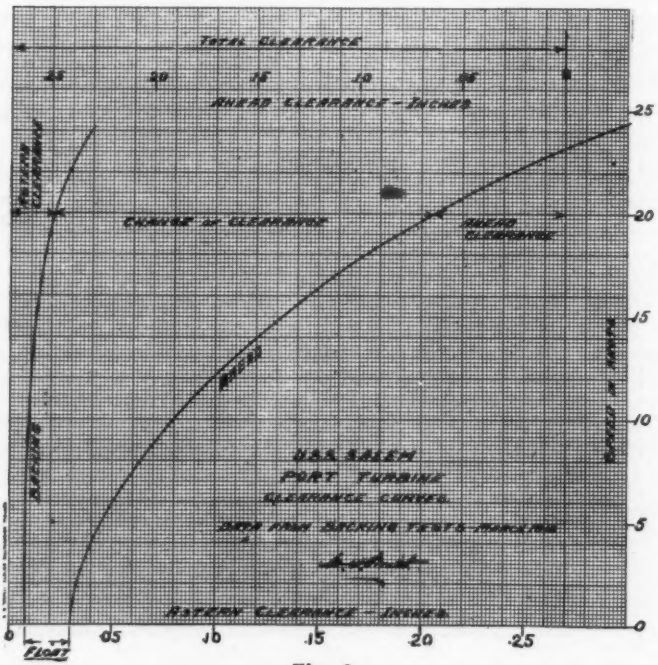


Fig. 3.

sufficient for ordinary conditions. When all boilers are connected steam can be supplied for the full backing power, and all nozzles on the backing end should be opened.

Steam Seals.—The steam seals must be watched and adjusted to suit the conditions, particularly when steaming at high power. If the glands are not properly sealed, air will be drawn in and the vacuum will drop.

Operating.—In changing speed or reversing, an excess of steam should be admitted until the desired speed has been attained, then the operating valve can be closed down as required.

In reversing there is considerable inertia to be overcome, and, in order to make the turbine respond quickly, the operating valve should be quickly opened as wide as necessary. When the turbine is reversing at the required speed this valve should be closed down the required amount.

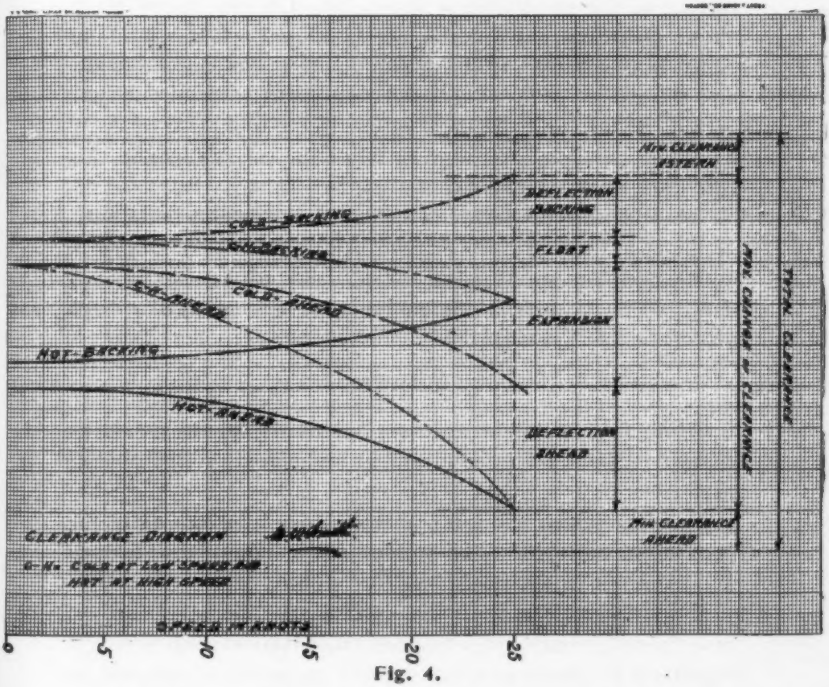


Fig. 4.

When a "stop" signal is received, the turbine is brought to rest by admitting steam for the opposite direction.

In operating the turbine, the number of nozzles that are open on each end and the steam-chest pressures should be kept in mind. The main steam, vacuum, gland and oil pressure should also be observed.

The temperature of discharge water and the speed of circulating pump must be watched, particularly when a full-speed signal is received.

If the steam pressure varies considerably, which frequently happens when steaming with reduced boiler power, the auxiliaries must be watched, particularly the oil pumps, which are liable to stop with a sudden reduction in steam pressure.

There is no difficulty in opening steam into both ends of the turbine at the same time if necessary. On one occasion the ahead operating valve could not be closed when a backing signal was received. The backing valve was opened wide, and the excess of backing power under these conditions reversed the turbine. As the ship was going alongside of dock, a number of signals were given while the turbine was in this condition, all of which were obeyed.

When handled properly the turbine works quickly and easily. If the clearances are properly adjusted there is no danger in going from full-speed ahead to full-speed astern—no serious damage can result from the handling, as it is practically "fool proof" in this respect.

UNDERWAY.

Steam Pressure.—At ordinary speeds the steam should be carried at about 250 pounds, and at high speeds at 275 pounds per gauge.

Nozzles.—Open enough nozzles to give approximately the revolutions required. The nozzles in use should be wide open and those not in use should be closed tightly.

Steam-Chest Pressure.—Enough nozzles being opened to give the revolutions approximately, the steam-chest pressure is regulated to give the exact number required. It should be carried about 25 pounds lower than the steam pressure, and be kept constant. Under ordinary conditions a change of about 2 pounds will change the speed 1 revolution.

Regulation.—At low speeds the revolutions can be regulated to within 1 revolution and at high speeds to within 3 or 4, or a little closer than 1 per cent.

Steam Seals.—Keep the pressure constantly adjusted to about 3 pounds. When the gland is under a vacuum, crack the drain valve and open the steam as necessary; do not open the drain valve wide and waste steam. When the gland is under a pressure, close the steam valve and open the drain valve as necessary.

See that the discharge to the siphon is closed and that to the fourth stage open.

When the speed is changed it is usually necessary to adjust the ahead seal.

Drains.—See that all the stage-drain valves, steam to siphon, and discharge to condenser are closed.

Warming Steam.—See that the valves at both ends are closed tightly.

Operating Valves.—See that the astern operating valve, balancing steam and by-pass are closed tightly. Feel astern steam chest occasionally to detect leakage.

Circulating Pump.—Regulate the speed by the temperature of the discharge water, which should be about 90 degrees F.

With hot injection water and high speeds it is necessary to carry a higher temperature, but with cold water and low speeds it is desirable to carry it lower.

Dry Vacuum Pump.—Adjust governor to give about 55 revolutions per minute. Set air valve in mid position, or where it will give the best results. If the compression is too high the release should be earlier, and *vice versa.*

See that the air cylinder and valves are lubricated properly. Use only special cylinder oil for this purpose.

See that the water service is running properly and that the cylinder and valves are running cool.

See that both steam valves are wide open, otherwise the governor will not act properly.

Hot-Well Pump.—Adjust governor to give about 1,250 revolutions per minute. See that the proper number of nozzles are open—two for low speeds and three for high speeds. Oil the governor frequently and see that it is running properly. See that the steam and water seals are properly adjusted.

Always run exhaust into main condenser and see that the valve to the auxiliary exhaust line is closed. The reason for this is that this turbine runs much more economically under a high vacuum.

See that both steam valves are wide open, otherwise the governor will not act properly.

Note the clearance each watch.

Oil Service.—Examine sight boxes frequently and see that oil is flowing properly. Note temperature of return oil frequently, and see that it is not too high. Under normal conditions it should not exceed 130 degrees, and it should never exceed 160 degrees.

The temperature of the return oil from the bearing at steam end will be higher than that from the exhaust end, on account of the difference in temperature of the turbine casing.

Carry oil tank about three-fourths full. It should not be carried full because it will overflow in case of stopping. See that the cooling water is circulating properly. Drain water from bottom of oil tank every watch, being careful not to drain off oil; also be careful to close valve tightly to prevent loss of oil.

Water Service.—Test frequently and see that it is running properly through bearings, cooling tank, dry-vacuum pump, etc., and see that it does not become air bound in high places.

Main Bearings.—Give particular attention to the bearing at the hot end of the turbine.

Thrust Bearings.—See that they contain sufficient lubricant and that the water service is running properly through *all* collars. See that nuts of side rods are set up tightly. This does not apply to thrusts of later type.

Auxiliary Exhaust.—Adjust spring-loaded valve to the desired back pressure. Ordinarily it should be opened into the third stage, but when the pressure in this stage exceeds the exhaust pressure (in high-speed running) it should be opened into the fourth stage.

Condenser and Vacuum.—Every effort should be made to maintain a high vacuum, because the economy of the turbine largely depends on this.

It should be remembered that the effective vacuum is that of the exhaust chamber and not that of the bottom of the condenser, consequently the vacuum reading should be taken from the exhaust chamber. There is little difference between the two at low speeds, but at high speeds it exceeds 1 inch.

The greatest loss in the operation of the turbines is that due to poor vacuum. Roughly, the steam consumption is increased 7 per cent. for a loss of 1 inch of vacuum from 28.5 inches, or about 1,500 horsepower at full power.

It has been found that impaired vacuum is generally due to the turbine glands not being properly sealed.

The most accurate vacuum gauges should be used, and these gauges should be calibrated frequently. Mercury gauges are more reliable and accurate than the spring type, which have not given very good results, but care must be taken that no mercury can get to the condenser.

Backing Turbine.—The pressure in the backing turbine with the ahead turbine in use should be the same, or slightly greater than, the pressure in the exhaust chamber. Steam leaking through the astern operating valve, the warming-up steam valve or the astern gland, will cause an increase of pressure in the backing turbine. The drain of the first reverse stage should be left open.

Clearance.—The clearances should be taken each watch and entered in the log. The clearance should not be less than 0.05 inch under any running conditions.

Economy.—The following points should be emphasized: (1) Carry high vacuum; (2) do not carry low steam-chest pressure; (3) work nozzles at full capacity; (4) do not allow steam to leak into backing turbine; (5) run auxiliary exhaust into turbines; (6) keep stage drains closed; (7) do not waste steam through gland drains; (8) run exhaust from hot-well pump turbine into main condenser to obtain the increased economy due to high vacuum.

Comparative Efficiency of Turbines.—When running together, the same number of nozzles should be required, provided the steam-chest pressures and the vacuum in the ex-

haust chambers are the same. By comparing the turbines in this way it can easily be determined whether there is any material difference in their steam consumption.

Priming.—Priming has little effect on the turbines, but is dangerous on account of the auxiliaries. On one occasion the boilers primed excessively for a considerable time, but there was no effect on the turbines other than to slow them down slightly.

STANDING BY.

When stopped and standing by for signals the oil pumps should be stopped to prevent loss of oil.

When standing by to get underway on short notice, the following should be done: Stop oil and dry-vacuum pumps, slow down circulating pump, open warming-up steam to turbines, open drains, and start siphon. When standing by for several hours, throw in turning gear and move rotor the same as when warming up.

FULL POWER.

Open all nozzles and carry steam-chest pressure at 275 pounds. When more steam is being generated than the turbines will take, open by-pass to second stage, but keep pressure steady at 275 pounds. The main turbines will give no trouble as long as the auxiliaries run properly. They run as easily at full speed as at low speeds.

SHUTTING DOWN.

Before stopping, shift the auxiliary exhaust from the turbine to the condenser.

When the order to secure is received, proceed as follows: Shut steam off glands; close operating valves tightly; open drains; close nozzles; shut down oil pump; do not close valves on oil system; drain water from cooling tank; cover bearings; shut down hot-well pump, leaving exhaust to condenser open; keep dry-vacuum pump running for about two hours to dry

out main turbine and hot-well pump turbine; keep circulating pump running slowly until turbine has cooled down a little.

In general, securing is the reverse of warming up. In securing, the auxiliaries should be drained, lubrication stopped, etc., according to usual practice.

CARE AND OVERHAULING.

When turbines are not in use the interiors should be kept dry and free from corrosion. There has been little trouble in doing this. The turbines retain their heat for a long time, which thoroughly dries them out after steaming. Sufficient oil is carried over with the steam to form a good protective coating on the interior surfaces. All valves leading to turbines should be tight so that water will not leak into them.

Some trouble has been caused by shafts rusting in the glands. This corrosion causes the rough surface of the shaft to cut the carbon packing, which wears away rapidly, is difficult to keep tight, and requires frequent overhauling. It is a good plan to oil the glands with cylinder oil and graphite after steaming.

Oil should be pumped through the bearings at least once a week when lying in port to prevent corrosion. The turbines and the auxiliaries should be turned daily according to usual practice.

The auxiliaries should be cared for the same as the auxiliaries of reciprocating engines, except as noted below. The interior parts of the turbines and auxiliaries should be inspected frequently. The best way to learn the conditions inside is by actual examination.

Casings.—The plates for examination on the exhaust chamber and underside of the casing should be removed and the inside of casing examined at least once a quarter. It will generally be found that the turbines are dry and well protected with a coating of oil.

Foreign Substances.—When any part of the turbine is open, the greatest care should be exercised to prevent foreign substances getting in. For ordinary repairs, as a rule, the tur-

bines should not be left open over night, or for any considerable length of time.

Steam Strainers should be examined every six months.

Glands.—The carbon packing should be removed, cleaned and refitted once in three months. The ends of the segments should be dressed down until there is from one-sixteenth to one-thirty-second inch clearance between the ends. This will allow for the ends coming closer together as the segments wear.

The springs of the segment holders should be replaced if they are corroded or weak. Spare springs should be carried in stock for this purpose. In the hot-well pump turbines the steel were replaced with brass ones, which have given better service. The shaft should be cleaned, and, if rough, it should be dressed smooth.

Oil Strainers.—The oil strainers should be removed and cleaned after each run.

Steam Chest.—The nozzle valve-stem packing should be kept in good condition. About once in six months the steam chests should be removed and the nozzles examined.

Vacuum Joints.—All joints, stuffing boxes, etc., subjected to vacuum should be made perfectly tight. The packing of movable joints should be examined frequently and should be set up tight or renewed as necessary to keep the joints tight. Other joints should be painted or cemented over all parts of the joint through which air can pass—over edge of gasket, around bolt heads, nuts, etc. This coating should be fairly thick and its surface unbroken. When such joints, after drying, are subjected to vacuum they should be examined and, if the surface of the coating is found drawn in and broken, more coating should be put on, or, if necessary, the joint should be remade. With joints cemented in this way, it is possible to tell by examination whether they are tight, which is not possible with ordinary joints. For this purpose asphaltum paint has given good results.

To make frequent inspection easy the joints should not be covered. Lagging is not necessary for parts under high vacuum.

Operating Valves.—The operating valves should be overhauled once in six months. The valves should be removed, cleaned, ground in, the rings fitted, etc. When necessary, the rods should be adjusted so that the valves will seat properly.

It is important to keep the astern operating valve tight to prevent leakage into the backing turbine while steaming.

Main Bearings.—The caps should be lifted and the journals examined once in six months. If the bottom brass is damaged and has to be replaced with a spare, it may require several days. The shaft and rotor must be jacked up and the brass driven out.

There is practically no wear on the bearings under ordinary circumstances. However, the vertical position of the shaft should be measured occasionally to determine whether the rotor is properly centered in the casing. If there is appreciable wear, new brasses should be put in or the bearing should be lined up with shims.

If the bearing is burned out or worn considerably, the diaphragm bushings will be worn down, which will increase the leakage. The bearing should not be allowed to wear down more than 0.02 inch. One-tenth inch wear on the bearings and the consequent wear of the diaphragm bushings increases the steam consumption about 8 per cent.

Thrust Bearing.—The clearance or "float" of the thrust bearing should be measured when checking up the turbine clearance. When it is necessary to adjust the clearance—the fore-and-aft position of the rotor—the nuts of the side rods are slacked back and the bearing drawn forward or back the desired amount, which is measured with feelers. Care should be taken to move each rod the same amount, otherwise the collars will be canted, and a hot bearing will result.

In adjusting the clearance in port, the position of the rotor is measured by inserting a feeler between the rotor and stator through a peep hole. If this cannot be done, it is measured by the clearance gauge.

Clearance.—The clearance gauges should be checked frequently and the scales set to the correct readings. The best

way to do this is to turn the rotor with the turning engine and move it axially in one direction until it strikes. Set the 0, or striking point, of the corresponding scale under the pointer and secure the scale. Move the rotor to the opposite striking point and set the other scale. Graduations on clearance gauge are shown in Fig. 5.

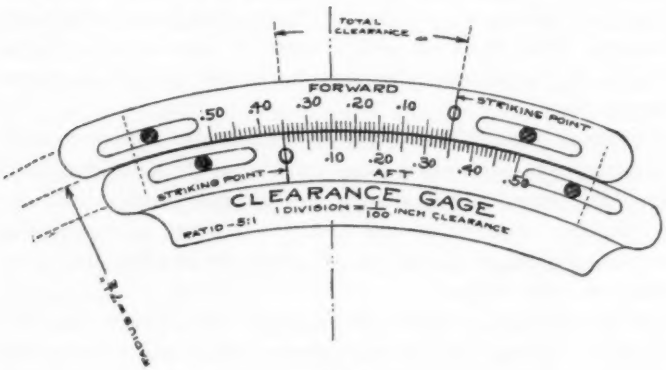


Fig. 5.

Ordinarily it is not desirable to disturb the adjustment of the rotor in checking the clearance gauges, and the following method is used: Remove plugs from peep holes and insert special feeler between rotor and stator at different points along turbine. Take smallest measurements on each side, which should be the same as the gauge readings. The rotor should be revolved and the same measurements taken at various points of the circumference.

No matter how accurately the clearance gauge is set, it cannot be depended on absolutely, on account of the inaccuracies of construction and the difference of expansion under the different conditions met with. For this reason it is not safe to run with too small a clearance by gauge.

Hot-Well Pump.—The turbine should be examined and overhauled the same as the main turbine. The ball bearings should be examined and cleaned occasionally. This pump

has been very satisfactory, and has given practically no trouble.

Dry-Vacuum Pump.—The engine should be cared for the same as any other engine. On account of the small clearance in the air cylinder it should be measured after overhauling the bearings. The valves should be taken down frequently, examined and refitted. The springs break occasionally and have to be renewed.

Oil Tank.—The cooling coil of the oil tank should be tested for leaks occasionally. The oil should be drained out and the tank cleaned at intervals of about six months.

In Dry Dock.—The propellers should be cleaned and highly polished. On account of the high speed of rotation every effort should be made to reduce the friction.

Lifting Casings.—The Navy Regulations require the casings to be lifted and the interiors of turbines examined once a year. This work can be best done at a navy yard, and requires about a month.

The exhaust pipes and upper parts of the casing are lifted out with a crane and landed on the dock. The rotor is lifted with the special lifting gear and all parts examined.

The greatest care must be taken to prevent dirt and foreign substances from getting into the turbine. The hatches should be roped off and the turbine carefully covered with canvas when no work is being done.

All joints are scraped down, and when the casing is put together great care must be taken to make the joints tight. Joints are made with manganasite, wire gauze being used for joints under high pressure.

When the casings were lifted in May, 1909, the following condition was found: Both turbines had been damaged by foreign substances and some of the buckets and nozzles were badly bent and battered. One row of buckets was completely closed. Rubbing had caused some of the shrouding to be worn away but had not injured the buckets.

The surfaces of the buckets and nozzles were in perfect condition as regards corrosion and erosion. The interiors of the

turbines were in good condition, and there was practically no corrosion.

Stage Valves.—Tests were made to determine the value of the stage valves originally fitted, and it was found that they had little or no effect on the steam consumption. For this reason, and on account of the undesirability of having fittings inside of the turbine that might get loose and cause damage, they were removed.

Turbine Oil.—A pure mineral oil should be used for forced lubrication. This should be free from acid; otherwise the bearings will be rusted.

Oil can be tested for the presence of animal or vegetable matter with a solution of borax and water. If such matter is present a white emulsion will be formed. The pure mineral oil will appear on top, the borax solution on the bottom, and the emulsion between.

Oil as a live element requires handling carefully to remain in good condition. Make-up oil should be added regularly to keep the full body as nearly constant as possible. This regular addition of new oil keeps it at about the same fluidity for the full period of use. If allowed to run too long without new oil being added, it will thicken and will ultimately have to be replaced as a whole.

No oil should be used which does not have a sufficiently high flash point, but the running temperature is well below the flash points of all oils used for this purpose.

The oil should have a fair body at the running temperature (about 120 degrees) and the viscosity should not fall off too rapidly at increased temperature.

If pure mineral oil is not used the constant circulation due to pumping will form a mechanical mixture which will cause the oil to emulsify and thicken, resulting in the system becoming clogged.

The flow of oil should be easy, and the tank should be sufficiently large for it to remain long enough to clear itself before passing into the succeeding cycle.

SUPPLIES.

The following special supplies should be kept on hand:

Main Turbines.—Two sets of carbon packing, carbon holders, springs for carbon holders, special packing for nozzle valve stems, two spare bottom brasses, and turbine oil.

Hot-Well Pump.—Two sets of carbon packing, carbon holders, springs for carbon holders (brass), spare balls for ball bearings, wheel-bucket segments, and intermediate bucket segments.

Dry-Vacuum Pump.—Springs for air piston and balance plate, valve rods, special cylinder oil.

Miscellaneous.—A good supply of aluminum paint, glasses for sight boxes, mercury for vacuum gauges.

Records.—Such special forms as are furnished for keeping the records.

Changes.—The following is a list of the principal changes made on the *Salem* since delivery to the Government:

1. Removed baffle plates and some tubes from upper rows in condensers to give a better vacuum in the exhaust chambers.

2. Installed hydraulic cylinders on operating valves to prevent excessive jumping in opening.

3. Installed exhaust connections from hot-well pump turbines to main condensers so that better economy could be obtained with the high vacuum.

4. Removed stage valves from turbines as noted above.

5. Installed double adjustable scales on clearance gauges so that the actual clearances of both sides of the rotor could be read directly from the gauges. (See Fig. 5.)

6. Installed by-passes to the second stages of the turbines to give greater overload capacity.

7. Installed guards on turbine shafts between glands and main bearings to prevent steam from blowing against bearings.

8. Installed oil-pressure gauges on main gauge boards.

9. Installed large (10-inch) accurate gauges for steam chests.

10. Changed gauge connections from gland steam pipes to

the glands, so that the actual pressures in the glands would be recorded.

11. Geared the revolution counters down 10 to 1, because they would not run satisfactorily at the high speed of the turbine.

12. Removed gravity oil tanks.

13. Removed shaft bilge pumps and connected water-service discharges to suctions of main circulating pumps.

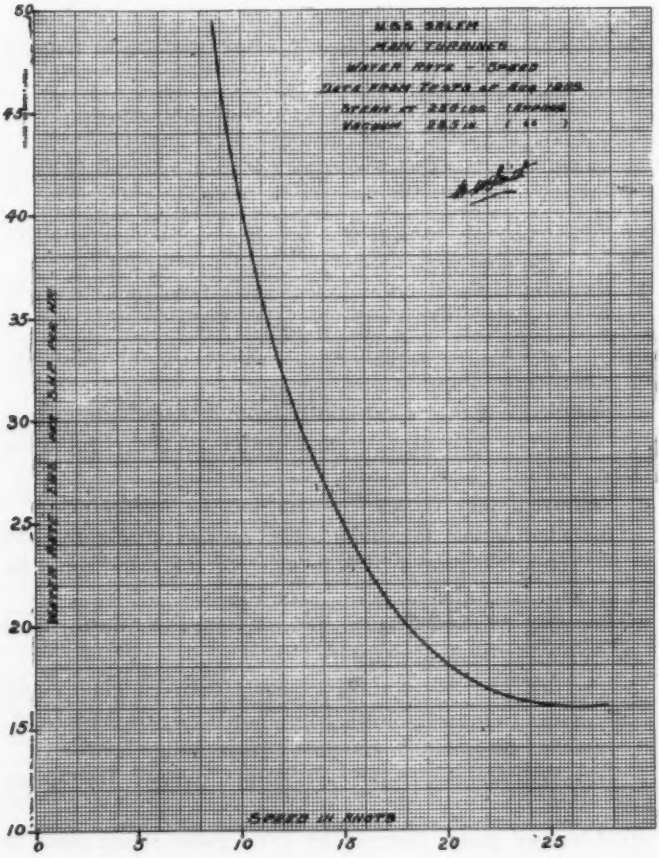


Fig 6.

PERFORMANCE.

Steam Consumption.—At full power the water rate per shaft horsepower is about 16 pounds and at 10 knots about 40 pounds. (See curve of water rates, Fig. 6.)

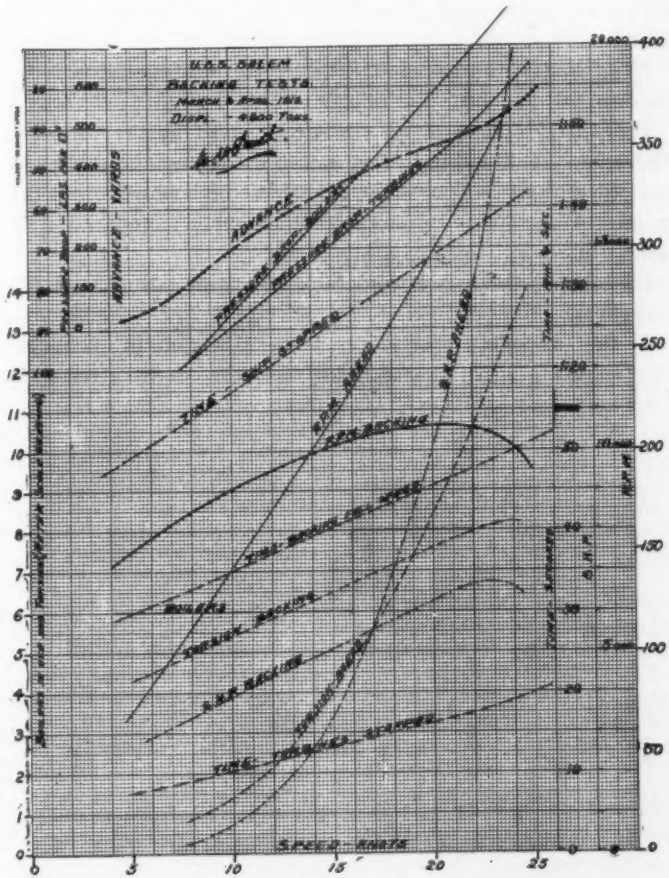


Fig. 7.

Vacuum.—There has been little difficulty in maintaining a high vacuum. With cold circulating water about 28.5 inches is carried and with warm circulating water about 28 inches.

Backing.—The backing qualities are excellent, and for this reason the ship handles unusually well. (See curves of backing tests, Fig. 7.)

Regulation of Speed.—The revolutions are regulated very accurately, as stated above. Compared with a reciprocating engine, the turbine can be regulated closer and the speed maintained more steadily.

Head Sea.—As compared with a reciprocating-engine ship, the decrease in speed, due to a head sea, is slightly greater.

Vibration.—There is no vibration caused by the turbines. There is a small amount of vibration caused by the screws at low power, and this increases slightly at high powers, but the greatest vibration is remarkably small when compared with vessels fitted with reciprocating engines.

In backing there is considerable vibration, caused by the propellers.

If the propellers are damaged the vibration is excessive. The vibration also increases with the wear of the after strut bearings.

SERVICE NOTES.

U. S. S. "BUFFALO."

PERFORMANCE DATA.

The U. S. S. *Buffalo*, formerly *El Sud*, of the Morgan line, is a single-screw transport used at present for service between San Francisco and Panama.

The notes which follow are compiled from reports made by Ensign Bruce R. Ware, Jr., U. S. Navy, Senior Engineer Officer.

Effect of Fouling on Speed.

The following table gives a good illustration of the results of fouling on coal consumption, revolutions, speed and power, and shows roughly the benefit derived from trimming one-third inch by the stern for each ten feet in length.

Runs	No. 1.	No. 2.	No. 3.	No. 4.
Date	Nov. 3, 1909.	Nov. 27, 1909.	Dec. 18, 1909.	Mar. 17, 1910.
Number of hours	24.0	24.0	24.0	24.0
Draught, forward, feet and inches	17–1	15–2	17–5	15–9
aft, feet and inches	21–8	20–4	23–2	20–4
mean, feet and inches	19–4½	17–9	20–3½	18–0½
Corresponding displacement, tons	6,000.0	5,300.0	6,400.0	5,400.0
Coal per hour, all purposes, pounds	5,017.6	5,998.7	6,737.9	6,431.1
Average revolutions per minute	64.08	67.12	69.2	65.83
I.H.P., main engines	2,501.2	2,562.5	3,087.8	3,015.5
all machinery	2,561.2	2,662.5	3,147.8	3,075.5
Pounds coal per I.H.P.	1.96	2.28	2.14	2.12
Knots per ton of coal	5.08	4.26	4.21	3.4
Average knots per hour	11.49	11.42	12.63	9.76
Slip of screw, per cent	18.1	21.6	18.2	31.7
Days out of dock	164.0	188.0	209.0	298.0

The dates of the above runs were selected when the conditions, force and direction of wind and sea were practically the same. All auxiliaries running in each case were the same. The condition of the bottom on runs Nos. 1, 2 and 3 was foul. On the fourth run it was very foul.

Effect on Speed of Cleaning Propeller Blades.

In the following table, on the run marked "*A*" the propeller was very foul, not having been scraped since the ship was in dock, May 24, 1909. On the run marked "*B*" propeller was clean, having been completely scraped by a diver before leaving port. The cost of this diver's work was $9.75.

Runs	"*A*"	"*B*"
Date	Mar. 16 to 19, '10.	Apr. 16 to May 4, '10.
Draught, forward, feet and inches..	15–09	15–11½
aft, feet and inches.....	20–04	21–02½
mean, feet and inches.....	18–00½	18–07
Displacement, tons.....	5,400.0	5,630.0
Coal per hour, all purposes, pounds.	6,431.0	6,766.0
Revolutions per minute.....	65.83	65.74
I.H.P., main engines.....	3,015.5	3,353.4
all machinery.....	3,075.5	3,426.0
Knots per ton of coal.....	3.4	3.57
Average knots per hour.....	9.76	10.8
Slip of screw, per cent.....	31.7	24.5
Mean days out of dock.....	298.0	330.0

Without taking into consideration the additional fouling of the ship's bottom for the time between the two runs, and using for a basis 3,500 miles, the coal required would be as follows:

For "*A*," 1,056 tons; For "*B*," 980 tons.

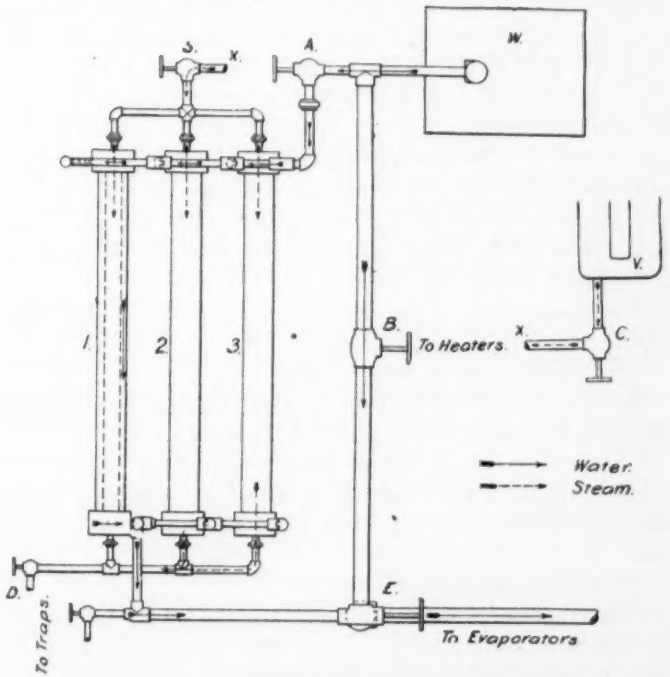
Thus the scraping of the propeller made a saving of 76 tons, and, at the average price of coal on the West Coast, this indicates a saving in expense of $634.30. The amount saved on the run from Panama to San Francisco was approximately $1,200.00.

Evaporator Feed-water Heater.

This vessel carries a total of about 950 persons, including officers, crew and passengers. The rated capacity of the evaporating plant, when forced, was 6,000 gallons for twenty-four hours. The ordinary capacity was about 4,000 gallons. In a desire to increase the efficiency so as to make sufficient water for the actual needs of all the men on board, an evap-

orator feed-water heater, as per accompanying sketch, was made and installed by the ship's force.

Referring to the sketch, 1, 2, 3 are three old steam radiators fitted with internal pipes, as shown at 1. "*V*" is the vapor line common to all three evaporators, and "*C*" is the shut-off valve to the heaters. Thus, by taking steam (heating element) from the common vapor line the heater can be run with all



or any one evaporator. This vapor passes through "*s*," as shown by broken arrows, through the star down the internal pipes in 1, 2 and 3, and traps off at "*D*" into the line to the ballast tanks, or after trimming tank. The feed water enters at "*W*" from the evaporator feed pump, passes through "*A*" into 1, 2 and 3, as shown by solid arrows, outside the internal steam pipe and out to the evaporators. By closing valves

"*C*," "*A*" and "*E*" the heater is cut out. By opening these valves and closing valve "*E*" the heater is cut in. Thus we have not only a feed-water heater but an extra distiller, with no additional drain on the boiler steam.

Previous to installing this heater the average water distilled per day was 4,810 gallons. With the heater running, and other conditions the same, it rose to 5,847 gallons, a gain of 20 per cent. It was also noted on the test trial that the pressure required to distil the same amount of water in the same time fell from forty pounds, without the heater on the coil of No. 1, to thirty-five pounds with the heater—a drop of three pounds on the coil of No. 2 and a drop of two pounds on the coil of No. 3. The average coal used per day for twenty days, with the heater running, making an average of 5,847 gallons per day, was 2.782 tons. The average before the use of the heater was 4,810 gallons and 3.221 tons per day. This shows a final gain of 21.5 per cent.

The temperature of the feed water was raised by use of the heater from 115 to 172 degrees F.°

A subsequent report is as follows:

Since the installation of the feed-water heater by the ship's force the following points have been noted:

(1) An increased shell pressure of five pounds.

(2) A much heavier deposit of scale, in proportion to the amount of water distilled, than when the heater was not in use.

(3) No corrosion was found.

The evaporators were run continuously under forced conditions from November 1, 1909, to May 5, 1910, scaled every two (2) weeks and on arrival here (Navy Yard, Mare Island, Cal.) stood a cold-water-pressure test of 105 pounds. During this period only one tube was renewed. From the results it is judged that the use of the heater has worked no injury to any part of the installation. With a draft of 800 men on board the gain of 21.6 per cent. in capacity and in economy was a necessity in order to produce sufficient fresh water for their needs.

THE LOSS OF THE "PLUVIOSE."

The recent deplorable loss of the French submarine *Pluviose*, which occurred through this vessel being struck by the passenger steamer *Pas-de-Calais*, has led to much speculation as to the exact sequence of events leading to the complete sinking of the vessel. The following is a translation of a communication to *Le Yacht*. The conclusions of the correspondent are interesting, and worthy of serious thought.

A photograph of the wreck of the unfortunate *Pluviose*, taken by a passenger on the *Pas-de.Calais* while she still remained above water, leads us to a very reasonable hypothesis as to the sequence of events leading to her final loss.

At the moment of collision the *Pluviose* was steering to the westward. Pierced in the motor compartment, and so violently that, according to the divers, a small boat could pass into the hull, she sank by the stern, reaching the bottom quickly in 21 meters.

The water quickly filled the motor compartment and passed into the engine compartment. These two compartments are separated by a bulkhead nominally watertight, but with several holes for pipes, of which one was about 20 cm. in diameter. The vessel rested on the bottom at the stern, with the bow above water, and after a time took up a position of equilibrium. The water, after having reached a certain level in the interior, was stopped by the pressure of the air. If nothing had been touched it is believed that the vessel would have remained thus for a long time—at least long enough to permit powerful tugs to tow her to the beach while dragging her by the stern. Unfortunately, the forward manhole was opened, as the divers have stated, following their operations. Water filled the remaining space and the vessel soon sank.

Why was this manhole opened? The photograph shows the bow above water, so that the forward manhole during the first moments of the accident was 4 meters under the water, supporting thus an outside pressure of .4 kilogram. Now, water having invaded the hull, had created in the part re-

maining empty a pressure of about .8 kilogram, due to the depth of immersion of the hole. The manometer, which only marked the difference in pressure, indicated then "surface," not having to support any outside pressure. It is very probable that, in view of this indication "surface," the unfortunate crew of the *Pluviose* believed that they were afloat and opened the manhole without fear.

It is very curious that a former commanding officer of the *Pluviose* had remarked since the wreck of the *Lutin*, which also opened her forward manhole, the anomaly of the manometer only indicating a difference in pressure instead of an absolute pressure, and attributed to this cause the loss of the submarine at Bizerte.

If the above hypothesis is correct it is probable that, if the manhole cover had been provided with a small glass plate, the crew of the *Pluviose* could have seen that they were below the surface.

The lessons to be drawn from this catastrophe are:

1st. The differential manometer should be replaced by an absolute pressure gauge.

2d. The bulkheads and doors which are *said to be* tight should be so in reality. It is remarkable that so much work should be expended on a bulkhead to adjust a door, making it perfectly watertight, and then there should be placed by its side a hole large enough to pass a man's body through. This condition is found not only in submarines but also in battleships.

3d. It seems advisable to fit a pane of glass in each manhole cover.

A fourth lesson may also be learned with profit, which is that we do not yet possess adequate facilities for lifting submarines of 400 tons, and some of them are of greater tonnage. Experiments made in the harbor of Cherbourg last March, on the condemned hull of the *Narval*, were not sufficiently conclusive. They should be repeated with an old hull representing more nearly the dimensions of a large submarine, and should be carried on to a point affording positive results.

Comment.

The editor of *Le Yacht* very justly observes: "As to the manometer incriminated, we do not think that it gave a false indication, for, from the first moment of the accident, the storage-battery compartment was filled, creating short circuits and extinguishing all the lights on board in the case of the *Lutin*, as in the case of the *Pluviose*."

"We agree with the conclusion that the bulkheads should be really watertight, and believe, with the author of this article, that with such bulkheads, and if the forward manhole had not been opened, the *Pluviose* would have remained in an inclined position in a state of equlibrium, with her bow above water, until help could arrive."

FOUNDERING OF A JAPANESE SUBMARINE.

On April 16th last, submarine *No. 6*, of the Imperial Japanese Navy, foundered while maneuvering in Hiroshima Bay, and all on board perished. She was commanded by Lieutenant Sakuma Tsotomu, and after the vessel was raised, the following letter of farewell from him was found in her conning tower. This remarkable document will be read with interest. The translation from the original, published in the Japanese press, appeared in the "Kobe Herald," and is as follows:

"Although there is, indeed, no excuse to make for the sinking of His Imperial Majesty's boat and for the doing away of subordinates through my heedlessness, all on the boat have discharged their duties well, and in everything acted calmly until death. Although we are departing in pursuance of our duty to the State, the only regret we have is due to anxiety lest the men of the world may misunderstand the matter, and that thereby a blow may be given to the future development of submarines. Gentlemen, we hope you will be increasingly diligent without misunderstanding (the cause of

this accident), and that you will devote your full strength to investigate everything and so ensure the future development of submarines. If this is done, we shall have nothing to regret.

"While going through gasoline submarine exercise we submerged too far, and when we attempted to shut the sluice valve the chain in the meantime gave way. Then we tried to close the sluice valve by hand, but it was then too late, the rear part being full of water, and the boat sank at an angle of about 25 degrees.

"The boat rested at an incline of above 13 degrees, pointing towards the stern.

"The switchboard being under water, the electric lights gave out. Offensive gas developed, and respiration became difficult.

"At about 10 A. M. on the 15th the boat sank, and under this offensive gas we endeavored to expel the water with a hand pump.

"At the same time as the vessel was being submerged we expelled the water from the main tank. The light having gone out, the gauge cannot be seen, but we know that the water has been expelled from the main tank. We cannot use the electric current entirely. The electric liquid is overflowing, but no salt water has entered, and chlorine gas has not developed. We only rely upon the hand pump now.

"The above has been written under the light of the conning tower, when it was 11:45 o'clock. We are now soaked by the water that has made its way in. Our clothes are pretty wet, and we feel cold.

"I had always been used to warn my shipmates that their behavior (on an emergency) should be calm and delicate while brave; otherwise we could not hope for development and progress, and that at the same time one should not cultivate excessive delicacy lest work should be retarded. People may be tempted to ridicule this after this failure, but I am perfectly confident that my previous words have not been mistaken.

"The depth gauge of the conning tower indicates 52, and

despite the endeavor to expel the water, the pump stopped, and did not work after twelve o'clock.

"The depth in this neighborhood being ten fathoms, the reading may be correct.

"The officers and men of submarines must be appointed from the most distinguished among the distinguished, or there will be annoyance in cases like this. Happily all the members of this crew have discharged their duties well, and I feel satisfied.

"I have always expected death whenever I left my home, and therefore my will is already in the drawer at Karasaki. (This remark refers only to my private affairs, and it is not necessary. Messrs. Taguchi and Asami! Please inform my father of this.)

"I respectfully beg to say to His Majesty, I respectfully request that none of the families left by my subordinates shall suffer. The only thing I am anxious about now is this.

"Please convey my compliments to the following gentlemen (the order may not be proper): Minister Saito; Vice Admiral Shimamura; Vice Admiral Fujii; Rear Admiral Nawa; Rear Admiral Yamashita; Rear Admiral Narita. (Atmospheric pressure is increasing, and I feel as if my tympanum were breaking.) Captain Oguri; Captain Ide; Commander Matsumura (Junichi); Captain Matsumura (Riu); Commander Matsumura (Kiku)—my elder brother. Captain Funakoshi; Instructor Narita Kotaro; Instructor Ikuta Kokinji.

"12:30 o'clock, respiration is extraordinarily difficult.

"I mean I am breathing gasoline. I am intoxicated with gasoline.

"Captain Nakano.*

"It is 12:40 o'clock."

—"Journal Royal United Service Institution."

*This is the name of another officer to whom the dying officer desired to be remembered.

THERMIT FOR SHIP'S REPAIRS.

The following is from a report by Lieutenant Paul Foley, U. S. N., commanding U. S. S. *Dixie*. The *Dixie* is fitted up as a repair and store ship and acts as parent ship for the torpedo fleet.

An interesting and at the same time convincing demonstration of the possibilities of thermit for general ship repair work is presented in a mold across a bearing surface of a fractured shaft recently made by the thermit process on board the *Dixie*, tender to the Atlantic Torpedo Fleet.

The shaft referred to belonged to the destroyer *Reid* and had been fractured during the final acceptance trial of that vessel. It was a single-crank solid-forged shaft, turning on a vertical axis at 900 revolutions per minute. It was 40 inches long and $2\frac{1}{2}$ inches in diameter with an axial hole $1\frac{3}{16}$ inches in diameter for internal oil circulation. The fracture occurred across a bearing surface $11\frac{3}{4}$ inches from the fan.

This then perfectly useless piece of metal was welded, restored to its original condition, and refitted in the *Reid* in 78 hours at a total cost of thirty-three dollars ($33.00). But aside from considerations of money economy, the operation was one of the most exacting ever attempted by the thermit process, within my knowledge.

The conditions to be satisfied were:

To recover and maintain the true axial alignment of the shaft, which was hollow.

To preserve the over-all dimensions of the shaft mathematically exact, one end of the shaft being fitted with thrust collars, and the fan end having no clearance whatever, the fan bearing being conical.

To unite the butts.

To avoid burning the metal adjacent to the weld, as the welded metal had subsequently to be turned down to a true bearing surface.

After welding to re-bore the axial oil hole.

The axial alignment of the shaft was recovered by re-

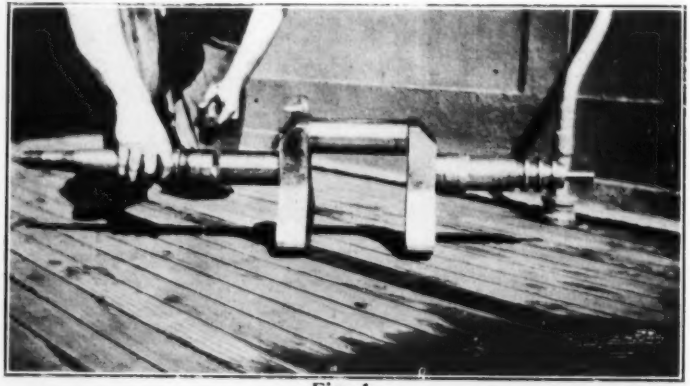


Fig. 1.

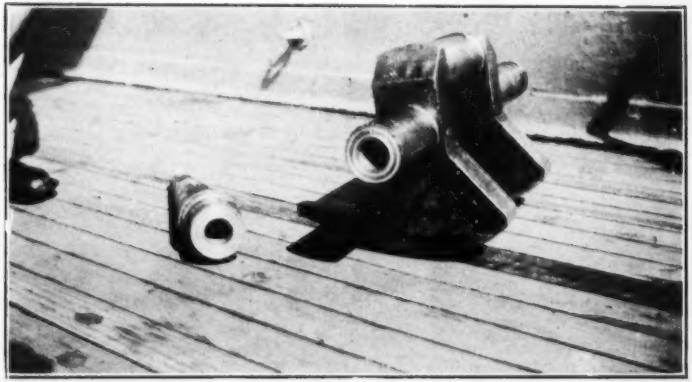


Fig. 2.

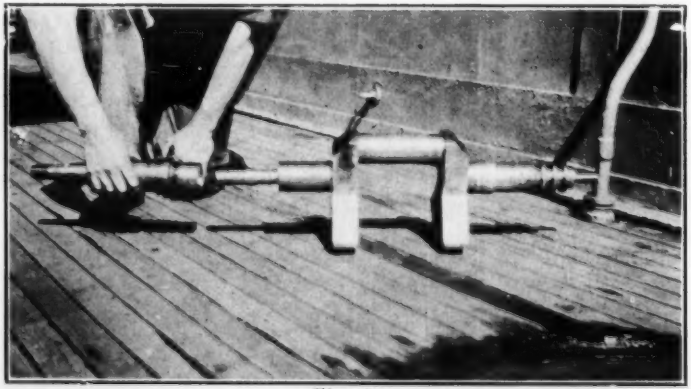


Fig. 3.

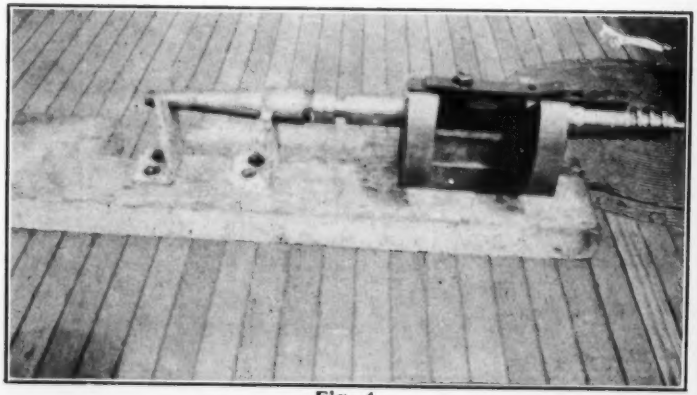


Fig. 4.

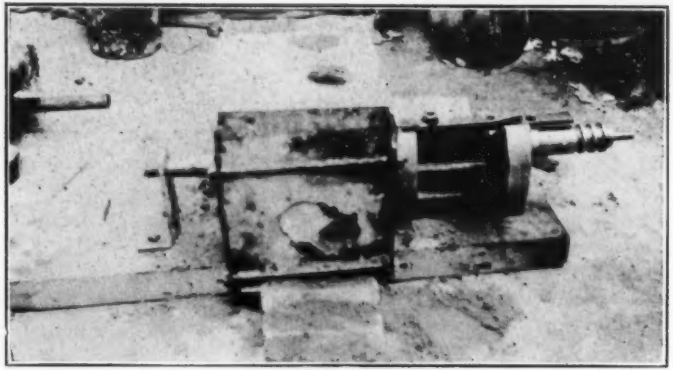


Fig. 5.

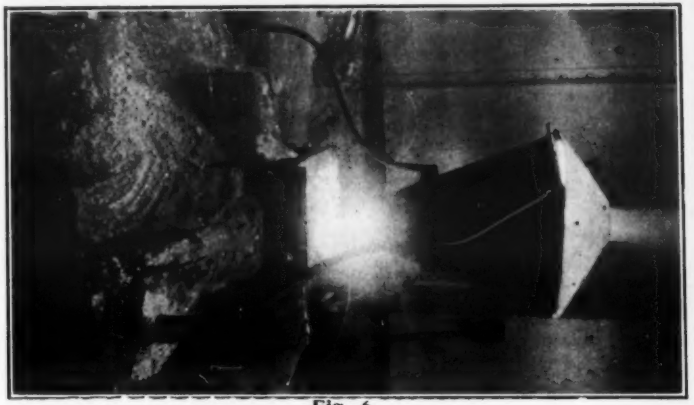


Fig. 6.

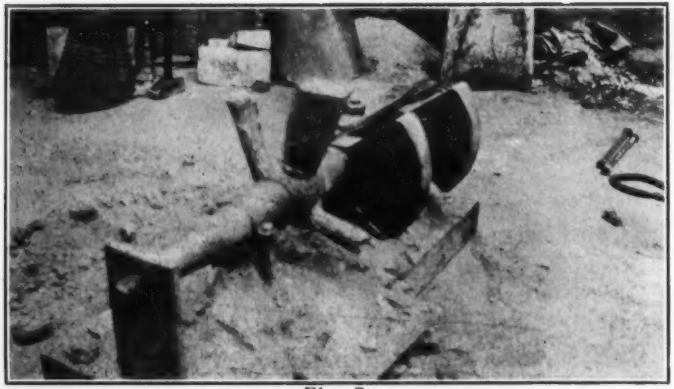


Fig. 7.

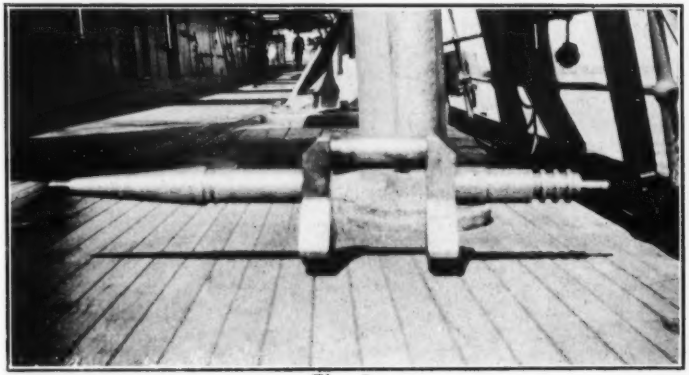


Fig. 8.

boring the axial hole of both broken parts truly concentric with the outer bearing surface and fitting a steel core between the two parts. The steel core was forced home in a lathe. It had an air channel cut longitudinally in its face to prevent any accumulation of air behind the core, which, under the intense heat of the thermit steel, might have exploded and disrupted the shaft. This part of the operation required fourteen hours of careful work.

Sufficient metal to allow for the thermit steel was turned off one of the butts before forcing the core home, and after the true length of the shaft had been re-established on the core by means of a steel tram, the shaft was securely clamped to a plank of three-inch ash, and the metal box for the mold set up around the fractured parts. This operation required seventeen hours. It was not hurried.

The thermit mold differs from that in ordinary use in that it is in one piece. Copes and drags are not used. The first stage in mold construction consists in forming about the butts to be united a collar of yellow wax molded to the exact shape that it is desired to have the thermit steel. In ordinary cases this collar would be left. In this case it was necessary to turn it down to a bearing surface. The second stage consists in improvising an iron box about the butts and filling it with green sand and fire clay. Wooden plugs are used to form the gate and riser, and a small piece of pipe led from the wax collar downward and out. This is subsequently plugged with a baked core. The third stage consists in preheating by means of a special gasoline torch. Preheating serves the triple purpose of melting out the wax, baking the mold and heating the sections to be united. The operation is considered completed when sections have obtained a bright red heat. This usually takes three hours. When preheating is complete the thermit crucible is set up over the mold and the thermit is placed therein and is ignited; this operation takes less than five minutes.

Thermit is a mixture of finely-divided aluminum and iron oxide. When ignited in one spot the combustion so started

continues through the entire mass without supply of heat or power from outside. It produces superheated liquid steel, and superheated liquid slag, aluminum oxide. The reaction produces a temperature of 5,400 degrees F. in less than 30 seconds. When this superheated liquid steel is poured upon the parts to be united it dissolves the metal which it comes in contact with and unites it to form a single homogeneous mass. Unless, however, the parts have been previously heated to a bright red, the incoming metal will exert a chilling effect and prevent successful fusion.

In this particular operation discussed there were used twenty pounds of thermit powder at twenty-five cents per pound, and two pounds of punchings at three cents per pound.

The mold was broken down five hours after pouring. The subsequent operations—machining and re-boring—required thirty-four hours. There being no off-set drill available for re-boring from the crank web end, a smaller axial hole was bored from the outer end and plugged. When finished the shaft was in absolute alignment, and the bearing restored to its original diameter and finish.

The various operations will be more clearly understood from the accompanying cuts, which are from photographs. The cuts are numbered according to the sequence of operations on the shaft.

The shaft was refitted in the *Reid* and is now in use, the *Reid* having steamed 1,000 miles thereunder.

There is nothing explosive about the thermit reaction, and no danger is incurred in storing or handling the material, owing to the fact that it requires the temperature of liquid steel to ignite it. Ignition can only be obtained by means of a magnesium flash-light powder.

SHIPS.

AUSTRIA-HUNGARY.

It is learned from a reliable source that the government of Austria-Hungary intends to create a fleet of sixteen *Dreadnoughts*. The construction of the first four, two of which have already been laid down, will be pushed with the greatest activity, followed by twelve others.

The government will present to the parliament a program of construction and scheme for raising the necessary funds to provide for these ships. Their life is fixed at eighteen years, after which they are to be replaced. The organization of this new fleet is to be based on that of Germany.—"La Vie Maritime."

BRAZIL.

The battleship *Sao Paulo*, sister to the *Minas Geraes* has recently completed her acceptance trials, the results of which are given in the following table, for which we are indebted to "Engineering":

Mean Results of all Trials.

.....	48 hours' coal consumption trial.		30 hours' trial.		8 hours' trial.		Six full-speed mile runs, 250 lbs. boiler press.	Two mile runs, 280 lbs. boiler press.
	Whole trial.	Mile runs.	Whole trial.	Mile runs.	Whole trial.	Mile runs.		
	1910.		1910.		1910.		1910.	
Date	May 21, 22, 23.		May 24, 25.		May 28.		May 30.	
Draught, forward, feet and inches.	25-02		25-00		24-05½		24-08	
aft, feet and inches.....	25-05½		25-03½		25-01½		24-10	
mean, feet and inches....	25-03½		25-01½		24-09½		24-09	
Displacement, tons.....	19,645		19,460		19,133		19,110	
Mean I.H.P.....	2,403	2,383	16,067	17,377	21,484	22,355	25,517	28,645
revolutions..... ...	67.15	67.52	127	129.8	138.7	140.85	145.59	150.85
speed, knots.....	...	10.61	...	19 853	...	20.99	21.231	21 623
Boilers in use..... ...	5	5	18	18	18	18	18	18
Coal consumption, lbs. per I.H.P.	1 8	...	...	...	1.42	...	...	...

The *Sao Paulo* was built and engined by Messrs. Vickers Sons & Maxim, who also furnished the engines for the *Minas Geraes,* which was built at Elswick. The following are the particulars of both vessels:

Length over all, feet	543–00
between perpendiculars, feet	500–00
Breadth, molded, feet	83–00
Depth, molded, feet and inches	42–03
Draught, feet	25–00
Displacement at 25 feet draught, tons	19,280.0
Speed at full power, knots	21.623
Indicated horsepower (I.H.P.)	28,645.0
Normal coal capacity on 25 feet draught, tons	800
Total bunker capacity, tons	2,360
Radius of action, determined on 48-hours' trial at 10.6 knots, sea miles	12,913.0
Armament:	
12-inch, 45-caliber guns	12
4.7-inch 50-caliber guns	22
3-pounder guns	8
Armor:	
Broadside	9-inch, 6-inch and 4-inch cemented steel.
Protective deck, inches	0–02
Main bulkheads, inches	0–09
Gun-barbettes, inches	0–12
Forward and after athwartship bulkheads, inches	3 and 4
Machinery—Engines, triple-expansion, four-cylinder type.	
Diameters of cylinders, high-pressure, inches	0–39
intermediate, inches	0–63
low-pressure (two), inches	0–73
Stroke, feet and inches	3–06
Total cooling surface of condensers, square feet	24,000.0
Boilers (eighteen) of Babcock & Wilcox type:	
Heating surface, square feet	58,370.0
Grate area, square feet	1,686.0

In the gunnery trials which followed the steaming trials, all the guns in one broadside, ten 12-inch and eleven 4.7-inch, were fired simultaneously, the weight of metal thrown being about 4 tons. The effect was to heel the vessel about 5 degrees, but no structural damage was done. Two 12-inch guns were leveled and fired directly over a lower turret in which officers were stationed, who experienced no discomfort.

A third battleship, to be called the *Rio de Janeiro,* has been

ordered from the Elswick yard of Sir W. G. Armstrong, Whitworth & Co., Limited. The new battleship will be 655 feet in length overall by 92 feet beam, and will have a draught of 26 feet, and a displacement of 32,000 tons. Her armament will consist of twelve 14-inch guns, fourteen 6-inch guns, fourteen 4-inch guns, three 6-pounder landing guns, six machine automatic guns, and three 18-inch under-water torpedo tubes. She will not only be very much larger than the *Minas Geraes* and *Sao Paulo*, but will also be bigger by a good deal than any battleship in, or at present in course of construction for any other navy. The machinery will be of the improved turbine type, to be manufactured by Vickers, Sons & Maxim, Limited, and will drive four screws and give a speed of 22½ knots. The approximate cost is understood to be £2,900,000, and the vessel is to be completed in two years.—"Engineering."

Floating Dock.—A new 22,000-ton floating dock of the self-docking type has just been completed by Messrs. Vickers, Sons & Maxim and delivered to the Brazilian Government for service at Rio de Janeiro. The following are some of the leading dimensions:

Over all length of dock over 25-foot platform, feet and inches	550–06
pontoons, feet and inches	500–06½
central section, feet and inches	165–03
each end section, feet and inches	167–07½
width of dock, feet and inches	136–00½
depth of pontoon at center, feet and inches	18–03½
foot of walls, feet and inches	17–09½
length of side walls, including docking lands, ft. and ins.	450–06
height of side walls above pontoon deck, feet and inches	45–06
and pontoon, feet and inches	63–03½
width of side walls at base, feet and inches	18–00½
top, feet and inches	12–06½
Clear width, feet	100–00
Maximum draught of vessel, feet	30–00
Height of keel blocks, feet	4–00
Depth of water required to dock vessel of 30-foot draught, including 6 inches for troughs and 5-inch clearance, feet and inches	53–03
Freeboard of side walls, feet and inches	10–06

Destroyers.—The official full-speed trial of the Brazilian destroyer *Sergipe,* built by Messrs. Yarrow & Co., took place on July 8th. The speed obtained was 27.676 knots on the mile and 27.605 knots during a continuous run of three hours' duration. The contract speed was 27 knots.

The *Sergipe* is the last of the ten destroyers ordered by the Brazilian Government from Messrs. Yarrow. The first seven vessels are now in commission in Brazil, the eighth has just left the Clyde and the ninth has been handed over to the Brazilian authorities and will shortly leave for Rio.

The dimensions of the Brazilian destroyers are: Length, 240 feet; beam, 23 feet 6 inches. They are propelled by two sets of four-cylinder reciprocating engines 8,000 horsepower collectively, balanced on the Yarrow, Schlick and Tweedy system. Steam is supplied by two double-ended Yarrow boilers of the latest type.

GREAT BRITAIN.

Liverpool.—His Majesty's cruiser *Liverpool,* built by Messrs. Vickers Sons & Maxim, Limited, has completed her steam trials, and realized a mean speed of 26.171 knots over the measured mile, the designed speed being 25 knots. The vessel was completed within sixteen months of the laying of the keel. The full-power trial was of eight hours' duration, within which period six runs had to be made over the measured mile. On these runs the revolutions of the four propellers averaged 512.7 per minute, while the shaft horsepower was 24,718, and the speed of the ship 26.171 knots. This power was easily maintained, and with great regularity, throughout the eight hours' run. The fuel consumption—oil and coal being used—was equivalent to 1.65 pounds of coal per shaft horsepower per hour.

The coal-endurance trials specified in the contract were a 22 hours' run at about 14,000 horsepower and an eight hours' run at about 18,000 horsepower, both trials being continuous. On the 22 hours' trials the power developed was 13,970 shaft

horsepower, and the fuel consumption was 1.57 pounds per shaft horsepower per hour. For the eight hours' trial the results were equally satisfactory, the shaft horsepower being 18,614, and the coal consumption 1.59 pounds per indicated horsepower.

The *Liverpool,* as the name suggests, belongs to the new "Town" class, of which nine are being built. The vessels are of the light, protective-deck type, with high speed, moderate armament, and great maneuvering power. They are 430 feet long between perpendiculars, 47 feet broad, and at 15 feet 3 inches draught have a displacement of 4,800 tons. The armament includes two 6-inch breech-loading guns—one on the forecastle and the other on the upper deck aft; ten 4-inch breech-loading guns—five on each broadside on the upper deck amidships; and two Maxim guns on the navigating bridge. There are also two submerged tubes for firing 18-inch torpedoes. Although the ships are unarmored, more than usually extensive arrangements have been made for ensuring the buoyancy of the ship in the event of the hull being perforated, by extensive subdivision of the interior; and there is extending right fore and aft a nickel-steel armored deck, sloping at the sides to well below the water line. Under this deck there are extensive longitudinal and transverse bulkheads, the machinery, for instance, being distributed over five compartments, while the boilers are arranged in three compartments. The machinery is of the Parsons type, with four lines of shafts. A high-pressure ahead turbine and the separate high-pressure astern turbine are arranged on one shaft in each wing compartment, while one low-pressure ahead turbine, in which is embodied the low-pressure astern turbine, is mounted on each of the inside shafts in one central compartment, which also contains the maneuvering gear for all the turbines. There is no separate cruising turbine, the principle being adopted of arranging for the admission of steam at several stages in the high-pressure turbine, according to the power desired. The economy obtained in the *Liverpool* would seem to indicate that this is a satisfactory arrangement.

There are four propellers—one on each shaft. The auxiliary machinery and condensers are accommodated in two separate compartments abaft the main engine room.—"Engineering."

New Destroyers.—The Admiralty has placed a contract with Messrs. Yarrow & Co., of Glasgow, for two torpedo-boat destroyers of special type involving many new features. In these vessels the Messrs. Yarrow will be allowed a free hand in the design of the machinery, and consequently the trials of these destroyers will be awaited with much interest. The forced-draft blowers on these vessels will be driven by Terry steam turbines of the type fitted in recent U. S. destroyers. Three of these turbines will be constructed by the Terry Steam Turbine Co. at their works in Hartford, and the remainder will be built by Messrs. Yarrow. Five sets in all will be placed on each vessel, each capable of furnishing 25,000 c.f.m. against 4 inches maintained air pressure.

SPAIN.

The rehabilitation of the Spanish Navy begins with a Royal Decree published in the "Diario Oficial" of April 23, 1908, inviting tenders for modernizing the dockyards at Ferrol and Cartagena, and building the following new war vessels: Three battleships of about 15,000 tons; four gunboats of 800 tons; three destroyers of 350 tons; twenty-four torpedo boats of 180 tons.

It was stipulated that firms tendering must be domiciled in Spain, where the construction was to take place, but such home firms were allowed to co-operate with foreign establishments, who would give their technical guarantee for the execution of the work.

The successful firm in the competition was the Sociedad Española de Construccion Naval, a Spanish corporation organized for this special work, and having the great advantage of the technical guarantees of three of the most important naval construction companies in the world. These are: Sir

W. G. Armstrong, Whitworth & Co., Limited, Messrs. John Brown & Co., Limited, and Messrs. Vickers Sons & Maxim, Limited.

The new battleships, which are to be named *España, Alphonso XIII* and *Jaime I*, are to be built at Ferral and are said to embody the following particulars: Length between perpendiculars, 435 feet; beam, 78 feet 9 inches; draught, 25 feet 6 inches; displacement, 15,450 tons; speed, 19½ knots; battery, eight 12-inch, twenty 4-inch Q.F.

The 12-inch guns are in four turrets, all on the same level. The forward and after turrets are on the center line, the remaining two being placed diagonally opposite, outboard. The arrangement permits all turrets to fire on either broadside, while three out of the four can fire directly ahead or astern. Ten 4-inch guns are placed on each broadside.

The sides are protected by 9 inches armor at the water line, which is carried up tapering to 3 inches at the upper deck. A bulkhead 1½ inches thick is carried down to the bilge for protection against torpedo attack. The turrets and barbettes have 10 inches armor and the 4-inch guns are behind the 3-inch armor on sides.

The machinery is of the Parsons turbine type, supplied with steam from Yarrow boilers. The turbines are arranged somewhat differently from early practice, the turbines of all four shafts being in series. There are one high-pressure turbine, one intermediate and two low-pressure turbines, the last being on the wing shafts. The high-pressure turbine on the port inboard shaft exhausts into the intermediate-pressure turbine on the starboard inboard shaft, and the steam passes from the latter to the two low-pressure turbines. The inboard shafts have also high-pressure astern turbines, exhausting respectively into a low-pressure astern turbine within the same casing as the low-pressure ahead turbine on the wing shafts. The arrangement of the ahead turbines, it is anticipated, will largely reduce the steam consumption at all powers, without in any way affecting the simplicity of maneuvering gear.

The first ship, *España*, will be delivered in about three years'

time, the second in four and a half years, and the third in six years.

At Cartagena the gun-boat *Recalde* is practically complete, while the second ship of this class—the *Laya*—is framed to the upper deck. The remaining gunboats—the *Bonifaz* and *Lauria*—will be laid down as the berths become vacant. These gunboats are 200 feet long by 30 feet in breadth and will displace about 800 tons. They will each be armed by four 3-inch quick-firing guns and two Maxims. Their designed speed is 13 knots, and the power of the reciprocating engines 1,100 indicated horsepower. Yarrow boilers working under natural draft will be fitted. The *Recalde* will be delivered complete next summer.

The destroyers, which will be named *Bustamente, Villamil* and *Requesens,* are 220 feet long by 22 feet in breadth, and will displace about 370 tons. Each will be armed with five 6-pounder guns and two 18-inch deck torpedo tubes. Parsons turbines developing 6,250 horsepower will be fitted, giving the vessels a speed of 28 knots. The boilers will be of the modified Normand type. Work will shortly be commenced on these destroyers, but as the first boat has not to be delivered for three years yet, no special pressure has been brought to bear in this direction.

The torpedo boats, of which there will be twenty-four, are 165 feet long by 16 feet 6 inches in width, and will displace about 180 tons. Parsons turbines, developing 3,700 horsepower, are being fitted, the steam being supplied by Normand boilers. The designed speed is 26 knots, and the armament will consist of three 3-pounder guns and three 18-inch torpedo tubes. The first three of these boats are to be delivered in about twelve months, and the work is well in hand.—"Engineering."

MERCHANT SHIPS.

Creole.—The Southern Pacific steamer *Creole,* the first large vessel to be fitted with Curtis turbines, has been recently refitted with reciprocating engines, by the Wm. Cramp & Sons S. & E. B. Co., Philadelphia. It is stated that the original machinery equipment, as described in the August, 1907, number of the JOURNAL, did not prove sufficiently economical to meet the expectations of the owners. In this connection, however, it should be stated that the Curtis turbines as installed in the *Creole* were the first large marine turbines of this type built, and much improvement in economy has been made since their construction. The new machinery consists of two vertical, inverted, direct-acting, triple-expansion, surface-condensing engines with cylinders 27¾ inches, 46½ inches, 79 inches diameter by 42 inches stroke, designed for 95 revolutions at 16 knots. The vessel's boiler equipment is the original ten Babcock & Wilcox water-tube marine boilers carrying a working pressure of 210 pounds above atmosphere with 50 degrees F. superheat.

On her first voyage with the new machinery the *Creole* left New York at 12:10 P. M. May 25, and arrived at South Pass, mouth of Mississippi River, at 7:58 P. M. May 28, 6 hours ahead of schedule. On account of dense fog it was impossible to run full speed up the Mississippi River. The vessel, however, arrived at her pier at 10:20 A. M. May 30, 10 minutes ahead of schedule. On the return trip this steamship sailed from New Orleans at 10:10 A. M. June 1, and arrived at her pier in New York at 3 P. M. June 5, 16 hours ahead of schedule time, thereby breaking all north-bound records between the two ports. The running time from bar to bar, south bound, was 103 hours 15 minutes, despite the fact that a delay of 2 hours 45 minutes was experienced between Scot-

land Light and Diamond Shoal on account of running at reduced speed in a dense fog. The running time between South Pass and Scotland Light, north bound, was 92 hours 20 minutes. The average speed, bar to bar both ways, was 16.55 knots per hour.

Later reports state that the *Creole* shows marked economy over the sister ship *Momus* of the same line. This is attributed chiefly to the Babcock & Wilcox boilers with superheaters fitted on the *Creole*.

OBITUARY.

Henry W. Fitch, Chief Engineer, United States Navy, Retired, who died on May 19, 1910, was one of the founders of this Society. He entered the Navy as a Third Assistant Engineer on May 3, 1859, and was promoted through the various grades, being commissioned a Chief Engineer March 4, 1871. He served with credit and distinction throughout the Civil War and, though retired in 1894, he was on active duty during the Spanish War, his last active duty. Chief Engineer Fitch was one of the most active of the officers of the old Engineer Corps. He was on duty at the Navy Department in the Bureau of Steam Engineering from 1877 to 1885 and was closely identified with the design of machinery for the first vessels of the new Navy. On several occasions he was a member of the engineering examining board, and was the inspector of machinery for the *Newark*, one of the first cruisers of the new Navy, whose machinery marked enormous advance over what had previously been used in the Navy. Chief Engineer Fitch will probably be remembered best because of his duty at the Naval Academy. The special course of instruction for the Engineer Division of Naval Cadets of the First Class was instituted under his direction. This course he made a success, and many officers who received their engineering training under his instruction will remember him through all their years with gratitude and affection. His advice was excellent; he corrected errors firmly and gently; he encouraged effort freely and generously. There are hundreds who feel that in his death they have lost a sincere friend.

JOHN K. ROBISON.

ASSOCIATION NOTES.

Lieutenant Commander U. T. Holmes, U. S. N., has been elected by the Council as Secretary-Treasurer to fill the vacancy caused on June 30, 1910, by the resignation of Lieutenant H. C. Dinger, U. S. N.

NEW MEMBERS.

The following membership has been added to the Society since the publication of the list of Members and Associates in the February number:

Members.

Burns, F. D., Lieutenant, U. S. N.
Caskey, G. L., Ensign, U. S. N.
Church, A. T., Ensign, U. S. N.
Decker, B. C., Commander, U. S. N.
Dodd, E. H., Lieutenant, U. S. N.
Doyle, S. H. R., Lieutenant, U. S. N.
Foote, P. W., Lieutenant, U. S. N.
Furlong, W. R., Ensign, U. S. N.
Graham, J. S., Lieutenant, U. S. N.
Hileman, J. L., Lieutenant, U. S. N.
Hill, Owen, Lieutenant, U. S. N.
Houston, V. S., Lieutenant Commander, U. S. N.
Johnson, C. H., 3d Lieutenant of Engineers, U. S. R. C. S.
Joyce, C. S., Lieutenant, U. S. N.
Kalbfus, E. C., Lieutenant Commander, U. S. N.
Kerr, Howard J., 3d Lieutenant of Engineers, U. S. R. C. S.
Krafft, Kurt W., 3d Lieutenant of Engineers, U. S. R. C. S.
Lackey, H. E., Lieutenant Commander, U. S. N.
Leary, H. F., Lieutenant, U. S. N.
Lewton, T. G., 1st Lieutenant of Engineers, U. S. R. C. S.

McCrary, F. R., Lieutenant, U. S. N.
McCullough, R. P., Lieutenant, U. S. N.
Maglathlin, W. C., 2d Lieutenant of Engineers, U. S. R. C. S.
Magruder, T. P., Commander, U. S. N.
Mannix, D. P., Lieutenant, U. S. N.
Mitchell, W. G., Lieutenant, U. S. N.
Norman, A. C., 1st Lieutenant of Engineers, U. S. R. C. S.
Orr, Henry C., Lieutenant, U. S. N.
Palmer, L. C., Lieutenant Commander, U. S. N.
Richardson, J. O., Lieutenant, U. S. N.
Roach, Henry C., 3d Lieutenant of Engineers, U. S. R. C. S.
Robinson, F. M., Lieutenant, U. S. N.
Smead, W. A., Lieutenant, U. S. N.
Stark, H. R., Lieutenant, U. S. N.
Strickland, G. B., Midshipman, U. S. N.
Williams, Hilary, Lieutenant Commander, U. S. N.
Woodward, C. H., Lieutenant Commander, U. S. N.
Wright, G. B., Ensign, U. S. N.

Associates.

Brown, Howard H., Editor International Marine Engineering, 17 Battery Place, New York City.

Cooper, L. T., Chief Machinist, U. S. N.

Gordon, W. J., Engineer Department, Vickers Sons & Maxim, Barrow-in-Furness, England.

Hermanrud, Torbjörn, 22 Edison St., Quincy Point, Mass.

Hibbard, H. L., Cutler Hammer Mfg. Co., Milwaukee, Wis.

Katzenstein, M. L., 65 East 81st Street, New York.

Kemble, Parker H., District Manager Edison Electric Illuminating Co., 360 Pearl St., Brooklyn, N. Y.

Leslie, S. I., The Leslie Co., Lyndhurst, N. J.

Martin, George R., Navy Yard, New York.

Martyn, David, 5903 Girard Ave., Philadelphia, Pa.

Metten, J. F., Chief Engineer Wm. Cramp & Sons S. & E. B. Co., Philadelphia, Pa.

Spear, George M., 855 N. Broad St., Philadelphia, Pa.

Stewart, Alex., Superintendent Motive Power, Southern Railway, Washington, D. C.

Thomson, Harold, Engineer Superintendent, Elswick Shipyard, Newcastle-on-Tyne, England.

Walton, Geo. E., 71 John St., New York.

Widell, Axel, Chief Engineer, M. C. N., Sverges Ångpanne Forening, Stockholm, Sweden.

Members and Associates are earnestly requested to bring to the notice of those eligible to active and associate membership, the advantages of belonging to this Society. Blank applications for membership will be sent by the Secretary-Treasurer on request.

With increased membership and with the finances of the Association in a prosperous condition, it is hoped to improve the standard of the JOURNAL by paying more liberally for original contributions.

A minimum rate of one dollar per page will be paid for original articles. This rate will be increased by vote of the Council for articles of special merit, particularly those giving the results of original research.

Translations of original articles of merit will be paid for at an appropriate rate, having in view the difficulties in translation. Foreign units should, when practicable, be changed to their English equivalents.